COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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High Pressures and Temperatures Planned for Marine Field

The Operation of Chain-Grate Stokers

Correlation of Steam and Electric Power in Industrial Plants

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COMBUSTION

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VOLUME ELEVEN NUMBER TEN

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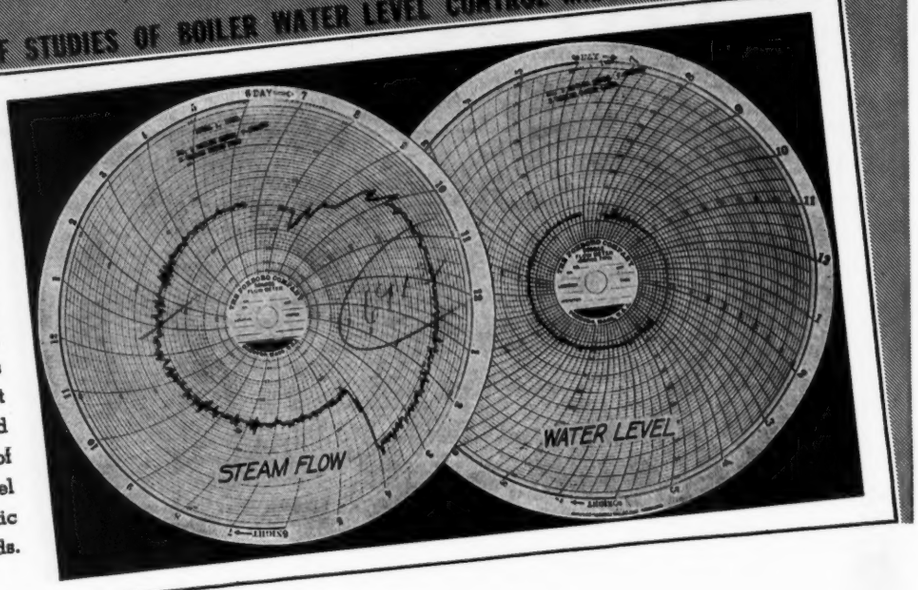

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EDITORIAL

Marine Practice Advances

Until recently marine engineers were often criticized for their apparent complacency in adhering to long-established practice, low steam pressures and temperatures, and their reluctance to follow advances in the stationary power field. Their defense was that conditions afloat were not comparable with those on land and that they were not warranted in taking a chance. This conservatism, to whatever extent warranted, now appears to be giving way to a disposition to follow certain well-tried practice in the stationary field. The widespread increase in steam pressure to 450 pounds in the merchant marine and higher pressures in naval work attests to this change in point of view.

Outstanding, in this connection, is the progressive attitude taken by the Technical Division of the U. S. Maritime Commission which, as pointed out by Mr. Schmeltzer elsewhere in this issue, is planning several 1200-pound installations, one of which will employ a steam temperature of 950 F. If these steam conditions prove feasible, as there is every reason to expect, the resulting improvement in economy should establish a new standard for certain types of vessels. The adoption of automatic combustion control and the placing of boilers and turbines in the same compartment are further evidence of following accepted stationary practice.

Opportunities for Combustion Engineers

In a paper before the Gas and Fuel Chemistry Division of the American Chemical Society at Cincinnati on April tenth, H. J. Rose, of Mellon Institute, discussed the opportunities for technical graduates in the field of fuel technology and combustion engineering. Despite the fact that this country's coal production is valued at about two billion dollars annually at the point of consumption, he observed that the coal-producing and processing industries do not appear to be getting their proper share of well-trained and aggressive young engineers such as are needed for the preparation and processing of coal, market development work, research, service advice and for ultimate advancement into executive positions. This situation is attributed partly to the former attitude of management in the coal industry and partly to the fact that the technical schools graduate very few men with substantial training in the utilization of solid fuels.

Aside from the coal-producing industry, there would appear to be excellent opportunities for more such men in the central station and industrial power plant fields, as well as among equipment manufacturers. In view of the extremely wide variation in coal characteristics, which have been intensified by recent advances in steam generating practice, combustion and slagging properties of coal assume paramount importance. While text-book knowledge and laboratory work do not suffice for the solution of these operating problems, still if the student were given more basic training in fuel technology instead of the nominal instruction in fuels, such as usually forms a part of the course in power generation, he would be better prepared to take up the practical work of fuel utilization and combustion in boiler furnaces.

Proposed Power Grid

Reports emanating from Washington indicate that the National Power Policy Committee, with the backing of the Federal Power Commission, is intent upon bringing about the construction of a vast high-tension grid system linking the principal cities and industrial centers east of the Mississippi. Into this transmission system, estimated to cost approximately 190 million dollars, both private and public utilities would be expected to feed. While the expense would be borne jointly by the Government and the utilities, a Federal Power Authority has been suggested for its construction and control.

The plan appears to be patterned somewhat along the lines of the British Grid although details are lacking as to how closely the latter might be followed. In the British system practically all the power companies deliver their output to the Grid from which power is delivered according to the demand and the most economical load dispatching. Additions to capacity and plans thereto are passed upon by the Commission.

It will be recalled that at the time the British system was conceived, operating practice among utilities in England comprised a large number of different voltages, frequencies, etc., with consequent very little interconnection of stations, and that the purpose of the Grid was largely economic.

But no such conditions exist in the United States. Not only are electrical frequencies and voltages standardized but most of the private plants already form components of systems and, in turn, many of the latter are interconnected as a means of meeting emergencies or for the economical exchange of power.

Proponents of a grid in this country are urging it, ostensibly as a national defense measure, probably in the expectation that public opinion and members of Congress are at present amenable to anything presented under the guise of defense, notwithstanding present extensive interconnection which many believe adequate to meet any likely defense demands. However, it is also proposed to make such a grid the outlet for St. Lawrence power as well as that from certain other Federal power projects. This, together with the frequently expressed desire further to extend Federal control over power supply, appears the main objective.

High Pressures and Temperatures Planned for Marine Field

The construction program of the U. S. Maritime Commission involves four high-pressure installations, one employing 1200 lb steam pressure and 950 F total steam temperature operating on the straight Rankine cycle and three using 1200 lb, 750 F steam with the regenerative reheat cycle. Details of these are given, as well as those of the large number of vessels employing 450 lb, 750 F steam. Many refinements in layout and equipment are included, leading to increased operating economy. Also, economic comparison is made between steam-driven and diesel-driven ships.

By J. E. SCHMELTZER

Asst. Director, Technical Div.,

U. S. Maritime Commission

ONE often reads of the American clipper ships which dominated our merchant marine a century ago. That these sailing ships were so successful in capturing and holding the shipping business of those days against foreign competition appears to have been due largely to their superior speed and their operating economy.

Taking a leaf from the record of these past experiences, the U. S. Maritime Commission has embarked upon a program of design and construction of a large number of cargo vessels to operate at speeds several knots faster than has been our previous practice with such vessels and with operating economies, as concern fuel expenditure per ton-mile, fully one-third better than the ships constructed twenty years ago. The present program involves about 200 ships of which 21 have been completed and 120 are under construction. Of these, over two-thirds are propelled by steam and the remainder by diesel engines. They include cargo ships, combination cargo and passenger ships and a few passenger liners.

Steam Conditions Advanced for Economy

To accomplish the desired operating economy it has been necessary to employ higher steam pressures and temperatures and many refinements in power plant layout, adapted to marine conditions, such as have proved most effective in land practice. While 450 lb steam pressure and 750 F total steam temperature have been adopted for most of the cargo ships, the Maritime Commission has given evidence of its progressive spirit by planning to employ steam conditions of 1200 lb, 950 F on one cargo ship and 1200 lb, 750 F, with the regenerative reheat cycle, on another cargo ship, for the American Export Line, and on two large 25-knot trans-Pacific liners of 80,000 shaft horsepower for which specifications have been issued and on which bids are now being asked. It is the opinion of the Commission's Technical Division that the experience to be gained with these high-pressure high-temperature installations will eliminate the necessity of advancing by steps through steam conditions in the intermediate range—a procedure that is usually costly.

The C-2 type of cargo ships, of approximately 9700 tons deadweight and 6000 shaft horsepower, designed for 15½ knots, employs 450 lb steam pressure and 750 F total steam temperature with cross-compound turbines and electrically driven auxiliaries. Each has two oil-fired water-tube boilers having a guaranteed efficiency of at least 87 per cent, and operated in all cases with automatic combustion control for which several types are being tried out. Two or three stages of feedwater heating are employed, according to whether economizers or air preheaters are provided. The feed system is closed and deaerating heaters are used for removing oxygen from the boiler feedwater. As is the practice with many of the later stationary installations, the boilers and the turbine are located in the same compartment in order to shorten the piping and to permit better observation by those in charge of operation.

Although designed for an overall fuel consumption of 0.60 lb of oil per shaft horsepower, the first ship of this class to go into commission did 0.545 lb on trial which is equal to 0.575 lb per shaft horsepower-hour for all purposes.

The C-3 type is somewhat larger and of 8500 shaft horsepower, with a speed of 16½ knots. It is one of these ships for which a study has been made for an installation to operate under steam conditions of 1200 lb, 950 F at the superheater outlet, operating on the Rankine cycle. A fuel consumption of better than 0.5 lb per shaft horsepower for all purposes is anticipated under such conditions.

The reheat cycle, as adopted with 1200 lb, 750 F for the one cargo ship and two liners mentioned, offers a decrease of 12 per cent in fuel consumption over 450 lb, 750 F in the case of the 8000-hp installation and approximately 15 per cent for the 80,000-hp plant. The smaller of these installations will consist of two high-pressure boilers each having one superheater of the

convection type and a separately fired reheater, located within the boiler setting. The only high-pressure steam line will be that from the superheater to the turbine throttle, which is a relatively short distance, all other piping being for 450 lb or less. Steam for all the auxiliaries will be supplied at 450 lb. The turbine will be of the triple-cylinder type having the high and intermediate sections in tandem.

Reheat and Rankine Cycles Show Same Performance Under Conditions Selected

The calculated performance for 1200 lb, 750 F with reheat is approximately the same as for 1200 lb, 950 F, without reheat, and it has the advantage of not requiring materials to withstand the higher temperature. Because of this it is not expected that such a plant will cost much more than that of a 450 lb, 750 F installation. Furthermore, the weight is not likely to be any greater for the cargo ship and will actually be much less per shaft horsepower for the two 80,000-hp liners. This is because of smaller high-pressure piping, a reduction in the size of turbine and a reduction in the size of certain auxiliaries due to the better steam consumption and to less fuel burned.

An additional advantage of the reheat cycle is ability to start the main units more quickly than if the higher steam temperature were employed, as the reheat can be cut out during starting. Its only purpose is to improve economy during steady operation.

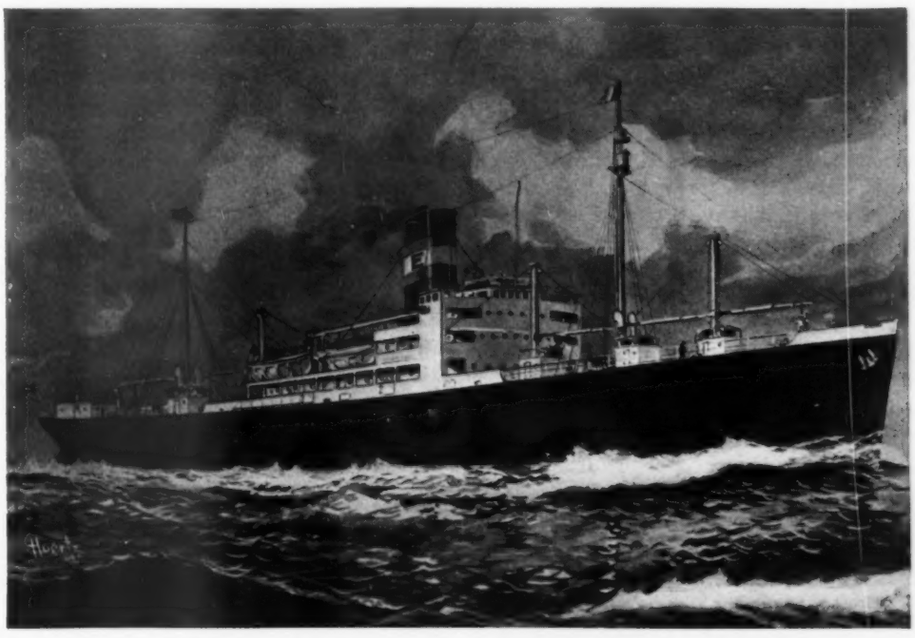
The large trans-Pacific liners will each have three evaporating plants with a combined capacity of 90,000 gal per day. These will use steam bled from the main turbines at 5 lb gage. In order to keep down the amount of makeup the soot blowers will employ air instead of steam. These will be operated continuously so as to avoid soot deposits on the decks such as would be likely with intermittent blowing. Air conditioning, refrigeration and lighting on these ships account for a very substantial auxiliary load.

Comparing the steam-driven ships with those propelled by diesel engines it may be pointed out that the former employ a somewhat lower grade of oil than the latter; so that, from the standpoint of fuel cost the economies of the two types are very close. However, recent developments in diesel engines, embodying reduced weight and space per horsepower through employment of higher speeds, have accentuated the problem of having a suitable fuel. Since the operator of a diesel cargo ship often has to purchase its fuel oil in foreign ports where he has no control of the quality, it is most desirable to employ engines that will economically burn oil of a wide range in quality if high maintenance and excessive operating costs are to be avoided.

Furthermore, in this country, the employment of improved refining processes for the extraction of gasoline is reducing the available amount of distillate such as is suitable for diesel engines and the domestic demand for distillate fuel is increasing with an accompanying increase in the market price—a factor that must not be neglected in considering the probable fuel costs for a diesel installation over its useful life, which may be considered twenty years. On the other hand, the fuel remaining after removal of gasoline and other distillates from the crude is usually suitable for burning under steam boilers.

Therefore, it is most important that motorships be propelled by engines capable of burning low grade fuel oil of a quality and price not too much in excess of that which can be burned in steam plants; otherwise the motorship, in general, will find it difficult to compete with steam.

In conclusion, it may be pointed out that while marked advances are being made in the marine field, through adoption of many practices and refinements that have been worked out satisfactorily on land, the marine engineer must ever keep in mind simplicity and reliability and the particular conditions imposed through operation at sea.



Cargo ship for American Export Line, to employ 1200 lb, 750 F with reheat

The Operation of CHAIN-GRATE STOKERS

By W. M. PARK, Service Engineer
Combustion Engineering Co., Inc.

A review of some of the fundamentals of chain-grate stoker operation to be observed when utilizing free-burning bituminous coals and lignite. The proper sizing of coal, range in draft for both natural and forced draft, coal distribution and segregation, the addition of moisture, control of air supply and overfire air are among the topics discussed.

A REVIEW of technical magazines and the literature of different stoker manufacturers or a study of existing plants reveals a large variety of designs not only as to stoker details but as to furnace construction, provisions for draft, draft control, coal- and ash-handling equipment, all of which may affect operation to various degrees. The operator can well afford to study these details so far as they are incorporated in his own installation. He should know how the functions and peculiarities of these may be inter-related to his chain-grate problem.

Size of Coal

Natural draft stokers usually have about 20 per cent air space in the grate and operate with 0.10 to 0.50-in. draft loss through the fuel bed. The former is about the least that can be used safely without danger of heating the grate excessively and nothing can be gained in the way of increased combustion rate by using over one-half inch draft above the fuel bed. Where the available furnace draft is low—say, 0.10 to 0.15 in., it is desirable to use coal containing little or no fines less than one-half inch size and a top size of one and one-quarter inches. Where a good draft is available, the favorite coal is a crushed product having $1^1/_2$ in. top size with the fines that naturally occur in crushing from lump or run of mine.

Forced-draft chain grates have seven to ten per cent air space so the increased fines incident to crushing to a smaller top size does not result in excessive siftings if the coal is fired with proper moisture content. The forced-draft design provides for as much as $1^1/_2$ to 2 in. drop through the fuel bed if conditions warrant this. Accordingly, these stokers readily handle coal crushed to $^3/_4$ or 1 in. maximum size. This smaller top size of fuel results in the coarser pieces burning out to a cleaner ash than is possible if crushed to $1^1/_2$ or 2 in.

Coal Distribution and Segregation

Where the installation of chain grates contemplates coal storage in overhead bunkers, it is worthwhile to insure uniform delivery across the bunker when filling and any spouts from bunker to stoker hoppers should be so designed that they will not segregate the coal, depositing the fines at the center and coarse coal at the sides of the stoker. Unless this precaution is taken, the fires will be inefficient because of their unevenness at various points across the stoker width. Traveling weigh larries and continuously swinging spouts are mechanical devices which practically eliminate segregation providing this fault does not exist in the storage bunker. Probably the worst conditions that can exist with spout delivery is where the bunker stands at one side of the stoker and the direction of the spout is such that it throws all of the coarse coal to the far side and the fines to the near side of the hopper.

Considerable ingenuity has been demonstrated by some firemen where these unfavorable coal conditions were not recognized and corrected by the management. Stoker gates have been altered to feed a deeper layer in the region where the coarse coal accumulated, thus causing the fuel bed to burn off to a more even length over the full width of the fire.

Tempering by Addition of Moisture

With few exceptions all bituminous coals can be burned to better advantage if their moisture content is increased several hours before burning. The evaporation of this extra moisture increases the superheated vapor item of the stack loss but this is more than offset by the improvement in CO_2 and the reduction of ashpit loss. Moisture on the surface of coal particles is detrimental, but that which is within the particle is of value and this penetration is effected to the desired degree if the coal is wetted 12 to 24 hr before firing. Where the tempering problem has not been provided for in the plant design, it sometimes proves impractical to give so much time for penetration of cold water. Moisture penetration can be greatly accelerated with some coals at a temperature of about 200 to 225 F. Fig. 1 shows a good arrangement for tempering in a stoker coal hopper using a series of $^1/_4$-in. pipes, spaced at about 5-in. centers and submerged in the coal. The top of each of these pipes enters a 2- or $2^1/_2$-in. header supplied with low-pressure steam of about 2 to 5 lb pressure. The header and pipes serve as a condenser and the water trickling into the fuel penetrates the particles. The larger lumps disintegrate into smaller pieces under the

action of the hot water and vapor, and this facilitates the contact between the streams of distilled gas and oxygen from the air supply. In the end, this makes possible combustion with less excess air and less unburned carbon in the ashpit refuse. With the arrangement shown, the manifold should have two or three inlets and should be set level so all pipes will receive the same amount of condensation.

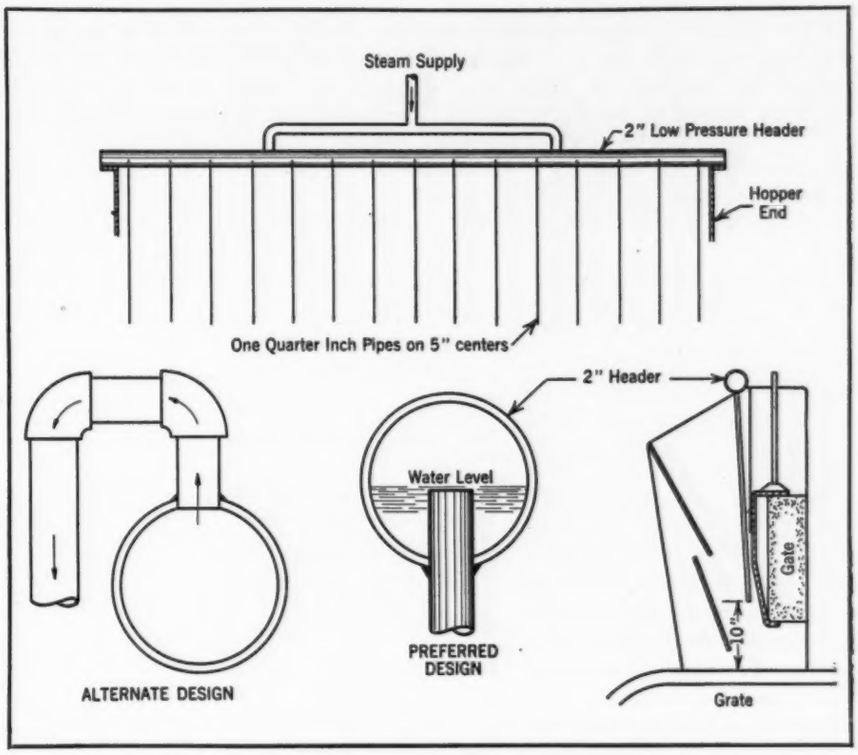


Fig. 1—Steam tempering arrangements

Control of Air Supply

Under this heading we shall not discuss control of the air in the usual sense that it must be carefully regulated to suit the fuel supply and the steam demand. Rather, we shall insist that the air enter the combustion zone in the way the designer intended, and indicate some of the devious paths through which short-circuits may occur. Instead of listing and describing these air leaks, a study of Fig. 2 will disclose the principal points for observation and correction. The measures to be applied will depend on the magnitude of the leakage, and the interest and willingness of the owner to make the necessary corrections. Since leaky ashpits are often ignored through misunderstanding of the consequences, several views in Fig. 3 are shown where some otherwise excellent designs of ashpits have failed to exclude air because of inattention to details. Stoker designers have given much thought to the means of constraining air to pass through the fuel bed and the firemen and maintenance men have no more important duty than to see that the designers' ideals are preserved.

The preceding remarks about air leakage are pertinent as affecting the efficiency of combustion with any kind of chain grate, either natural or forced draft. Aside from affecting efficiency, air leakage may seriously reduce capacity with a natural-draft chain grate. Numerous air leaks into the combustion chamber have an effect similar to opening a door of the furnace, in that it "breaks the vacuum" which draws combustion air inward against the fuel bed resistance. This effect is always most evident in a plant where a single boiler is required to handle all the load and ofttimes the elimination of air leakage will rejuvenate a sluggish, slow-steaming unit, thus greatly reducing the work and anxiety of the conscientious fireman.

Since the intensity of furnace draft is such an important item with a natural-draft stoker, it is well to consider some other things which affect the draft beyond the furnace defects. The areas of baffle openings in the boiler gas passages are proportioned by the manufacturer with due regard

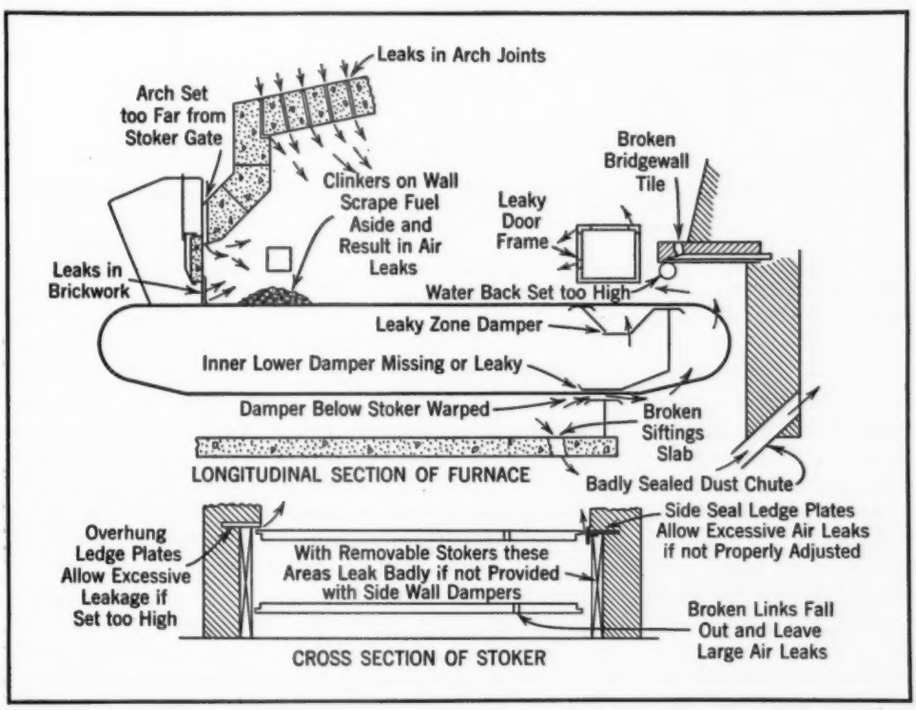

Leaks in Arch Joints
Arch Set too Far from Stoker Gate
Clinkers on Wall Scrape Fuel Aside and Result in Air Leaks
Broken Bridgewall Tile
Leaky Door Frame
Leaks in Brickwork
Water Back Set too High
Leaky Zone Damper
Inner Lower Damper Missing or Leaky
Damper Below Stoker Warped
Broken Siftings Slab
LONGITUDINAL SECTION OF FURNACE
Badly Sealed Dust Chute
Side Seal Ledge Plates Allow Excessive Air Leaks if not Properly Adjusted
Overhung Ledge Plates Allow Excessive Leakage if Set too High
With Removable Stokers these Areas Leak Badly if not Provided with Side Wall Dampers
Broken Links Fall Out and Leave Large Air Leaks
CROSS SECTION OF STOKER

Fig. 2—Sources of air leaks sometimes found in furnaces with chain-grate stokers

for the balance between efficiency and the cost of producing draft at the source. Baffles which have been disturbed for purpose of inspection, cleaning or tube renewals, should be carefully restored, particularly as to the original proportions, else they may reduce the furnace draft and the capacity. Accumulation of soot and ash in the boiler passes, in the breeching and at the stack openings will likewise reduce the draft over the fire.

Boiler dampers sometimes are attached to the shafts by "U" bolts and these or the damper-operating levers may slip and allow the damper to stand partly closed even though the external levers be set in their usual position. In one case under observation, a main flue damper located in a secluded passageway and held open by a counterbalance weight, became closed when this weight fell off. With the resultant loss of draft the coal ceased to burn properly, and the ashpits filled with burning coke which eventually damaged the grate links beyond repair. In addition to four new chains, the owner had to build a temporary header connected to several locomotives with their expensive rental and operating costs while the new grates were being delivered and installed.

If the draft at a stack is subnormal for its height and gas temperature conditions, it may require some pointing of masonry or patching or renewal of a metal stack. A good pair of binoculars or any spy glass will enable an observer to stand on the ground some distance from the stack and see any serious defects of this nature. Induced-draft equipment while more flexible than stacks, is subject to troubles detrimental to furnace draft and should be understood and maintained properly. To the writer's knowledge, one plant installed a single induced-draft fan to handle five boilers. Dispensing with the manufacturer's supervision, the owner set this three-quarter housed fan over a concrete rectangularly shaped duct, that is, without a scroll of any kind to continue the fan scroll. A lot of gas simply revolved continuously with the fan rotor instead of discharging. As a result, a very low intensity of draft was produced at the fan inlet and practically none at the natural-draft chain-grate furnaces until such time as the scroll was provided to insure positive discharge and normal draft.

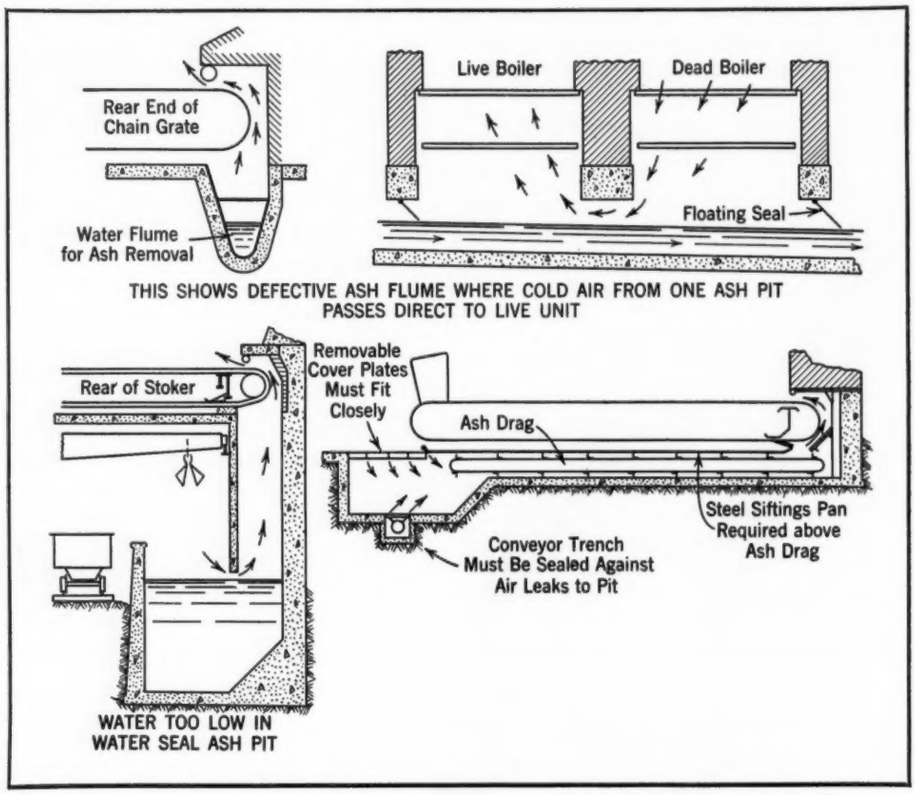

Rear End of Chain Grate
Water Flume for Ash Removal
Live Boiler
Dead Boiler
Floating Seal
THIS SHOWS DEFECTIVE ASH FLUME WHERE COLD AIR FROM ONE ASH PIT PASSES DIRECT TO LIVE UNIT
Rear of Stoker
Removable Cover Plates Must Fit Closely
Ash Drag
Steel Siftings Pan Required above Ash Drag
Conveyor Trench Must Be Sealed Against Air Leaks to Pit
WATER TOO LOW IN WATER SEAL ASH PIT

Fig. 3—Illustrating how air leakage may exist around ashpits and ash-removal equipment

Ignition

Ignition of fuel on the chain grate occurs partly by contact and partly by absorption of radiant heat. Considering the phenomenon of ignition more in detail, we shall discuss what happens to a lump of coal as it leaves the stoker hopper and enters the furnace. This particle receives heat to such a degree that it finally gives off gas by distillation. This gas ignites, at first possibly some distance from its source in the particle, but ultimately flashes back to the particle itself. It continues to distill gas, part of which burns directly at the surface of the particle (if oxygen can reach it), but most of it must remain as a hydrocarbon or carbon monoxide gas until it can find its quota of oxygen for completing the combustion.

Now, as to the source of heat which first warmed the particle and started its distillation, we must consider where the particle happened to be in the fuel bed as it left the coal hopper. If it is at the top surface, then it receives considerable radiant heat from the flame in the furnace and from the incandescent brickwork of walls, arches and the stoker fuel bed. This is sometimes called ignition by radiation. If, on the other hand, the particle enters the furnace in the lower part of the fuel bed, it receives its heat from combustion of adjacent particles which have previously ignited. For lack of a better term, the latter is often referred to as ignition by contact.

Since ignition necessarily precedes combustion, it is evident that the combustion rate or its ultimate result capacity is limited by the ignition rate. This important fact merits study of the line of demarcation between the ignited and unignited particles on a grate. Referring to the sketches of fuel bed sections on a chain grate,

Fig. 4, moving in the direction shown by arrows, if all particles in the bed could receive radiant heat at the same rate, then the ignition line would be as shown at *a*. However, since the top particles receive heat by radiation as well as contact, they ignite more rapidly than the lower layers and the normal ignition line is more nearly as shown at *b*.

If for the sake of experiment, it should be possible to refrigerate the lower part of the fuel bed, then its particles would not be raised to the ignition temperature as quickly as in *b* and the ignition line would be more like that indicated in *c*. Now this actually takes place if

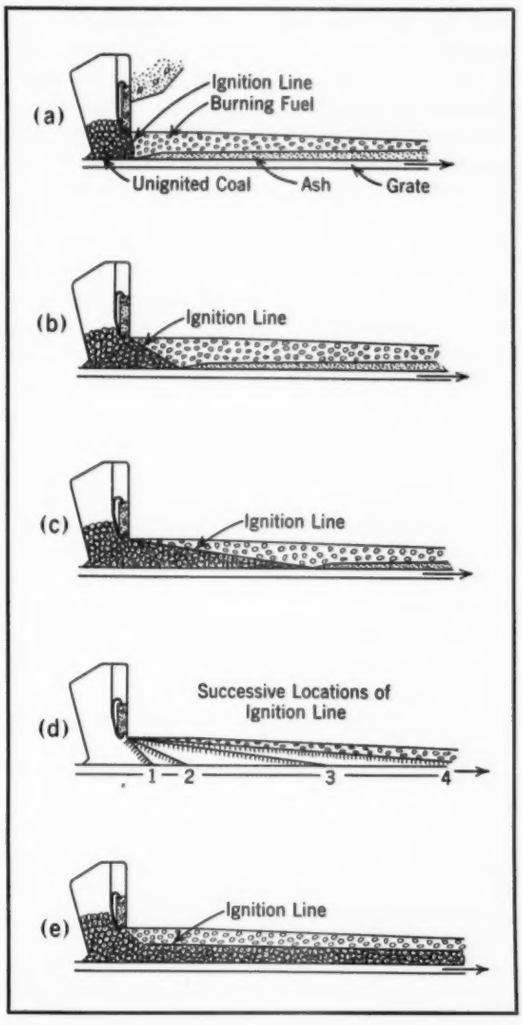


Fig. 4—Showing location of ignition line under normal and abnormal conditions

too much air is passed through the fuel bed at the ignition zone. It may occur either, due to using too much air pressure, or due to coarse coal in which more voids are present and therefore less resistance through the fuel bed.

"Radiation" and "contact" have been mentioned as two components of the ignition phenomenon. If now for further study of the problem we observe a stoker in operation and picture its fuel bed, it will be of interest to note the effect of grate travel. When the speed is increased sufficiently, the rate of entrance of new particles and the heat required to ignite them exceed that which can be applied by radiation and contact. As a result, the ignition line in the fuel bed advances in the direction of the grate travel, as to points 1, 2, 3 or 4 in sketch *d*. At first this gives no trouble, but since it gradually results in reduction of length of the active fuel bed, the heat release and flame radiation decrease until ultimately the entire fire is lost.

Noting this effect of excessive grate speed, it is natural next to try reducing the grate speed and thickening the fire, so as to feed the same amount of fuel. That procedure, however, ultimately leads to a fuel bed condition like that shown in *e*, where the radiation component may hold the surface ignition but the contact component cannot keep pace with the rate of new particle supply in the lower stratum of the fuel bed. The unignited lower layers of green coal ultimately run over into the ashpit. So it makes no difference whether we run the grate too fast or the fire too thick, the results are unfavorable because we have tried to feed and burn coal faster than it can be ignited.

An understanding of these principles and the ability

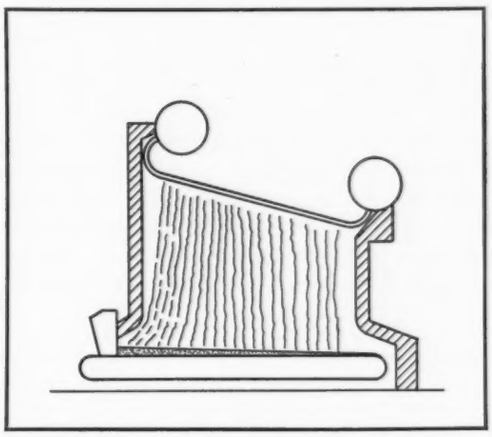


Fig. 5—Stratification of gas and air in the absence of arches

to recognize what is happening will help the operator when faced with ignition failure. When the trouble persists it may require a study leading to modification of furnace design, but the present discussion is confined to the operators' problem. Where there are several boilers in operation it is unusual to lose ignition on all furnaces simultaneously, so a failing unit can be slowed down until ignition improves and in the meantime its load can be distributed among the other operating units. Where the character of the load is such that the operator can predict the time of the peak, he can prepare for this by feeding additional coal to the furnace so it will be ignited and ready for the increased steam demand.

When ignition weakens on natural-draft stokers, it is well to stop the stoker and partly close the boiler damper until the ignition line gets back to the stoker gate. This reduces capacity somewhat but not as much as it will if the stoker speed is left unchanged. If there is only one boiler in the plant, there is no alternative to dropping the load until ignition is restored. On a forced-draft stoker it is less difficult to restore lost ignition be-

cause in that case the air is forced rather than drawn through the fuel bed and the boiler damper can be closed sufficiently to produce either a neutral or a slight plus pressure, and this always is effective in bringing the ignition point back to the stoker gate.

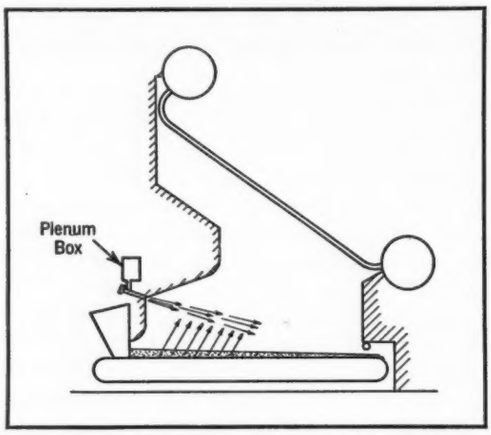


Fig. 6—Overfire air jets directed across strata of gas and air above fuel bed

Elsewhere we have dwelt at length on the importance of constraining air to pass *through* the fuel bed, since this is the way to insure the optimum of volatilization of the chain-grate fuel. At this juncture, however, it is pertinent to note that good volatilization is not necessarily followed by good combustion. This can best be illustrated by studying Fig. 5 which shows one type of chain grate furnace recently advocated to a limited extent. The unusual feature is that this furnace has practically nothing in the way of conventional arches. The front and rear furnace enclosures merely form seals with the stoker and the ashpit wall.

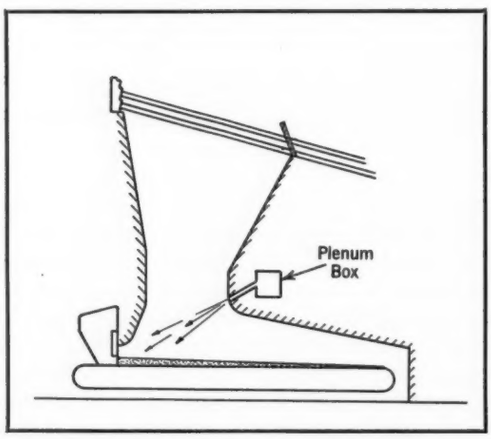


Fig. 7—Overfire air directed toward ignition point

Volatilized gas rises from the fuel bed and flows in parallel strata or streams. Unfortunately, however, one stratum may consist of combustible gas and the adjacent stratum of atmospheric air. There is nothing in the furnace to cause the strata to mix and it is too late to effect the combustion reaction after the gases have entered the relatively cool space among the boiler tubes. Such a furnace is likely to be sluggish in response to increase in steam demand with any kind of coal and impractical with low volatile coal. It will smoke badly and give high stack losses due to incomplete combustion.

As designers depart from this impractical arrangement and approach the conventional types, they still face the problem of stratification. Arches set at various angles and covering varying percentages of the grate length, by their reflected heat and the turbulence they create, all tend greatly to accelerate the combustion after volatilization.

If a chain-grate furnace is required to operate at only one rating, it is possible so to design the arches that they will give the turbulence necessary. However, since practically every furnace must operate over a considerable range of capacity, it is better to supplement the undergrate air supply with some overfire air jets. If it is done with arches alone, the maintenance may be excessive because of the small gas areas and high velocities required to insure turbulence. It is just as important to constrain the direction and flow of overfire air, as it is to direct properly the underfire air through the fuel bed. To illustrate this refer to Fig. 6. Note that the overfire jets are directed from a plenum box to flow across the stream of stratified volatiles and unmixed air arising from the grate. Since the overfire air is from a source where the static pressure is of far greater intensity than that through the fuel bed, it can actually cut through the hot gas and air strata, causing thorough mixing and prompt combustion. If there is a convenient means of preheating the overfire air, this adds to its effectiveness on the speed of reaction.

When the overfire air is introduced through the rear arch, as in Fig. 7, it advantageously distorts the flow toward the entering fuel. The resultant increase of heat liberation close to the fuel bed accelerates ignition down through the fuel bed as discussed elsewhere. It will be evident at once that this can greatly assist in maintaining ignition of excessively wet coal or low-volatile or high-ash fuel. Naturally, good judgment must prevail in this application to high-ash fuels, if the ash may have low fusing temperatures.

Before leaving this subject of overfire air, it may be pointed out that it is just as essential that the secondary source of oxygen be constrained in its entrance and its pathway, as is the case with the primary source through the fuel bed. To illustrate how one plant paid the penalty of only half doing the job and not carrying through the designer's idea, plenum box and overfire air nozzles were installed in the proper manner, but instead of connecting to a source of air supply and undergrate pressure, the plenum box end was left open to receive air by natural draft. Under this condition air left the nozzles at very low velocity, and because of its density and weight, it dropped like a cold blanket directly on the entering fuel, retarding the ignition and seriously reducing the furnace capacity.

Sometimes the overfire air is supplied by the main forced-draft fan, but a more positive control is possible if an independent fan be used. This should have a capacity for supplying ten per cent of the total air requirements at approximately 10 to 12 in. static pressure in the plenum box. It is better to have a larger number of small nozzles closely spaced, rather than a few nozzles widely spaced. The designer generally finds it necessary to give consideration to the width of a manufacturer's standard arch tile in establishing the spacing of the nozzles.

Correlation of Steam and Electric Power in Industrial Plants*

By G. G. HOLLINS, Chief Engineer
J. G. White Engineering Corporation

The author discusses the various factors that enter into a study of the most desirable heat balance, as dictated by prevailing steam and power demands, conducive to minimum operating costs in an industrial power plant. In this connection steam conditions, generated and purchased power, types and combinations of equipment, and steam versus electric drive for auxiliaries are considered.

ONE of the first considerations in the correlation of steam and electric power is the amount of power required and the amount of process steam that is used. It is obvious that in industries having a comparatively small use for steam the conditions are not favorable for low-cost power generation. With process steam used at low pressures and high-pressure steam delivered to a turbine exhausting at the low pressure, much can be done with a small quantity of steam, but even then the steam use should have a time load characteristic of about the same type as that of the electric power load. In other words, steam for building heating is not generally useful in generating low-cost power throughout the year. There are exceptions to this, as in the case of a plant that generates excess power in cold weather and trades it with a utility for power in warm weather.

An unbalanced condition existing between steam and electric power demands is not a problem if the steam demand at low pressures exceeds that necessary for making the required electric power by a noncondensing turbine-generator taking steam at the boiler pressure and exhausting or extracting at the pressures required by the process, as the turbine-generator can under these conditions supply the electrical demand without wasting steam. The problems arise when the steam use is insufficient all or part of the time for this simple method of power generation.

Possible Combinations To Be Considered

The economical supply of steam and electric power to an industrial plant requires a study of the following methods of steam and power production:

Generation of steam and purchase of power.
Generation of steam for process and power generation with steam condensed or wasted or the use of internal-combustion engines for power.
Generation of steam at sufficient pressure to generate power and to use all the extracted and exhaust steam for process.
Combinations of the preceding.

In regard to the first, if use of steam is small or too much out of phase with the electric power requirements and the cost of purchased power is low, it may be more economical to have a low-pressure steam plant sufficient for the process steam or heating requirements and purchase the electric power.

If, on the other hand, purchased power is expensive and the fuel cost low but still the steam use is not suitable for making power with an extracting or noncondensing turbine, the second method may be advantageous, by making the power with an engine of the uniflow or four-valve type, a straight condensing turbine or an internal-combustion engine.

The third method is the ideal heat balance of power generation for an industrial plant and is obtained when all the power required can be made by the steam passed through a turbine or turbines before going to the process, and the simplest arrangement is when the turbine exhausts at the pressure required for process. When two process steam pressures are required, the automatic extraction turbine is useful as it will supply the process steam requirements at the respective pressures within its load limitation.

The automatic extraction feature increases the cost of the machine appreciably in the smaller sizes and also increases the straight-through steam rate. For two regulated extracted steam pressures the double-automatic extraction turbine is used. This means another item of extra cost and further reduction in the efficiency of the turbine.

When there are engines, steam-driven pumps or air compressors in the plants producing an excess of exhaust steam, it may be advisable to use a mixed-pressure turbine into which this exhaust steam is introduced and expanded to a lower pressure to generate power. The mixed-pressure feature may be applied to an automatic extraction turbine, either to introduce the steam through the extraction opening or through another opening.

It is frequently not possible to have an ideal heat balance or perhaps it would be better to say that the ideal

* From a paper before a joint meeting of the Metropolitan Sections of the A. S. M. E. and the A. I. E. E., New York, March 21, 1940.

heat balance may be approached but not frequently obtained. If obtained, it is often not maintained through the faster growth of the electric power load over that of the steam load.

When making additions or improvements in existing plants, there is usually equipment which is too costly or too new to be replaced and this means that the new equipment must be a compromise with the ideal design. For instance, if the existing plant contains a good boiler plant operating at, say, 400 lb pressure, there is a serious question as to the advisability of installing a single new boiler at a higher pressure. Or if the existing plant contains a number of direct-acting steam pumps operating only a few hours per day, probably it will not be practical to change them to motor-driven pumps. Of course, in actual cases each must be investigated.

Also, space and distance factors in an existing plant often seriously limit what might be done for better performance of the power plant. Process steam may be required too far away to use low-pressure steam and the distance for returning condensate may be too great. Following is an example of the increasing of power production with low cost equipment rather than making the investment required by more economical equipment:

Typical Examples

In a plant using exhaust steam at 30 lb pressure from a noncondensing turbine-generator, the quantities being about 50,000 lb per hr in winter and about 25,000 lb per hr in summer, a small condenser was installed to take steam from the 30-lb system and maintain the cold weather steam demand on the turbine. The supply of steam to the condenser is controlled by a valve which opens when the process steam demands diminish and closes when the process demands increase.

A condenser is used instead of venting the excess steam to the atmosphere, so the condensate may be returned to the boiler and keep down the percentage of makeup and the cost of water treatment. Also, the wasting of steam to the atmosphere is bad in its psychological effect on the operators, making them careless about reducing other losses. This would seem like a case for a low-pressure turbine but a study showed that with the total amount of excess steam available, there would not be sufficient return on the investment in a turbine. The cost of the fuel is low (11 cents per million Btu), which is the principal reason for the economy.

As an example of the benefit in economical power generation to be obtained with an automatic extraction turbine, the following case may be cited, some of the details being somewhat out of the ordinary:

An industrial plant had a 2000-kw noncondensing turbine taking steam at 385 lb and exhausting at 20 lb for process steam requirements. The power generated was not sufficient for the mill requirements and additional power was purchased. Much of the steam at 20 lb was used to heat water to a temperature of 120 F. The heating of this water could have been done with steam at a much lower pressure and more power obtained from the steam that had the same water-heating capacity.

To accomplish this a 3500-kw single automatic-extraction turbine-generator with a condenser was installed to operate with a vacuum of 27 in. The water to be heated is passed through the condenser to a warm-water storage tank, whence it is pumped to the departments using the warm water.

The automatic extraction opening on the turbine supplies the requirements for 20-lb process steam and the additional steam necessary to maintain the load goes on to the condenser and heats the water. Roughly speaking, the steam that goes all the way through the turbine to the condenser produces twice as many kilowatt-hours per 1000 lb of steam as that taken out at 20-lb gage.

The use of warm water is irregular and the circulation of cooling water in the condenser must be continuous and at a sufficient velocity for effective heat transfer, hence water from the warm-water storage tank is recirculated through the condenser with the cold water to replace that used from the tank. The proportion of recirculated water is regulated by a temperature-controlled three-way valve on the suction side of the circulating pump set to maintain the temperature of the outlet circulating water of the condenser. Actually, this valve comprises two 16-in. butterfly valves, one in each branch pipe, linked together and operated by a power cylinder. The water-storage tank has an overflow so that if there is an excess of warm water, it will drain back into the well from which the mill pumps take their supply. The general use of water is sufficient so that there is no likelihood of the temperature building up on the condenser.

As would be expected, the temperature of the heated water varies considerably. A supply of water must be maintained at all times in the warm-water pipe lines and a low-level float in the storage tank supersedes the temperature control so as to maintain a minimum water level. The variations in water temperature have not caused any difficulty as the maximum temperature is set so that some additional heating is still required at the point of use and if the water is below the nominal temperature, only more heating is required.

This system has been in operation two years and has worked very well. The turbine-generator has operated at an average capacity of over 90 per cent for the time in service, excluding Sundays. Besides the increase in electric power production, the reduction in the direct use of steam in heating water has reduced the percentage of makeup to the boilers from about 40 per cent to under 20 per cent. The boiler makeup is heated in a stage heater taking steam from an unregulated extraction opening at about 6 lb absolute.

Employment of Thermo-Compressors

Thermo-compressors are inexpensive and have a field in increasing the power production by increasing the pressure of low-pressure steam for use at a pressure which otherwise would be supplied through a reducing valve. They operate on the same principle as the steam-jet vacuum pump on a condenser, only entraining and compressing steam instead of air. The range of capacity of an individual unit is small as the entrainment ratio increases rapidly as the output of the unit is reduced by throttling the high-pressure steam.

With high-pressure steam at 385 lb, 575 F and low-pressure steam at 20 lb with the discharge at 60 lb, such a unit stops entraining steam at 70 per cent capacity and acts simply as a reducing valve at lower outputs. The average ratio with changing load can be improved by using several compressors operated in sequence by a pressure controller responding to the discharge pressure.

For the steam conditions mentioned and a set of four compressor units, the entrainment ratio will range between 2.8 to 1 and 4.1 to 1 for outputs from half to the full capacity of the set. The low ratio is that of one unit at full capacity and occurs when all the compressors are on full and the high ratio occurs when one unit is throttled to the point of no entrainment. With this arrangement there is a high ratio of 4.1 to 1 at 67.5 per cent capacity and another of 3.8 to 1 at 92.5 per cent capacity. The more the units in service, the lower are the upper limits of the ratio.

On a recent installation it was estimated that a noncondensing turbine installation to supply the same quantity of steam at 60 lb pressure would require over five years to pay for the additional investment by the value of the greater amount of power generated. As the turbine installed would cost over ten times that of the thermo-compressor installation and there was a possibility of the operating conditions changing, the lower investment was preferred.

Topping Units Not Always Justified in Industrial Plants

The subject of topping turbines has a popular appeal and often owners of industrial plants expect that a topping turbine will do much more to solve their power generation problems than is warranted by the facts. The conditions relating to a topping turbine in an industrial plant are quite different from those in a public utility plant. In the latter when topping is considered, there is usually some low-pressure turbine-generator equipment that is in reasonably good physical condition and a boiler plant that is old and of inefficient design. The plant as a whole cannot compete with a modern plant and it may be difficult to convince a public service commission that it should be maintained in the rate base. By installing a modern high-pressure efficient boiler plant and topping turbines, exhausting at a pressure suitable for the existing turbine-generator equipment, practically a modern plant is obtained of increased capacity and the lower pressure turbine-generator installation with its auxiliaries and circulating-water system obtains a new lease of life.

In the industrial plants, turbine-generators exhausting at steam pressures suitable for process requirements or operating existing equipment have been in use for some time but have not been thought of as topping in the sense the word is applied to public utility plants. Generally, in an industrial plant the steam requirements are such that a noncondensing turbine exhausting at high pressure does not best fit the conditions and the proper solution will probably be a combination of extracted and exhaust pressures. For instance, steam might be extracted at 150 or 200 lb for operating certain equipment and also extracted or exhausted at lower pressures for process requirements.

If an industrial plant is already using power generating equipment, taking steam at about 400 lb, conditions are generally not favorable for superimposing a turbine-generator exhausting at this pressure unless the power requirements are sufficiently large. This is because the initial steam conditions for a superimposed turbine would have to be about 1200 lb, 900 F and about 5000 kw is the smallest topping turbine-generator for such steam conditions that is commercially available at present. The quantity of steam exhausted from this size of machine at full load would be sufficient for about 7000 kw turbine capacity, operating between 400 lb and 20 lb and about twice as much capacity operating between 400 lb and two inches absolute pressure.

Pure feedwater is essential for high-pressure boilers and a plant using steam for process usually has a high percentage of makeup, too large for economical evaporation. This makes the water purification an important consideration and one requiring close attention. Furthermore, the average industrial plant operator is not trained for the satisfactory operation of boiler plants operating at high pressures and temperatures. An industrial does not have the same incentive as a public utility to experiment with high efficiency equipment. Also, it does not have the same amount of spare equipment. With an industrial, power is a necessary but secondary consideration and must not divert attention from the principal business of producing merchandise.

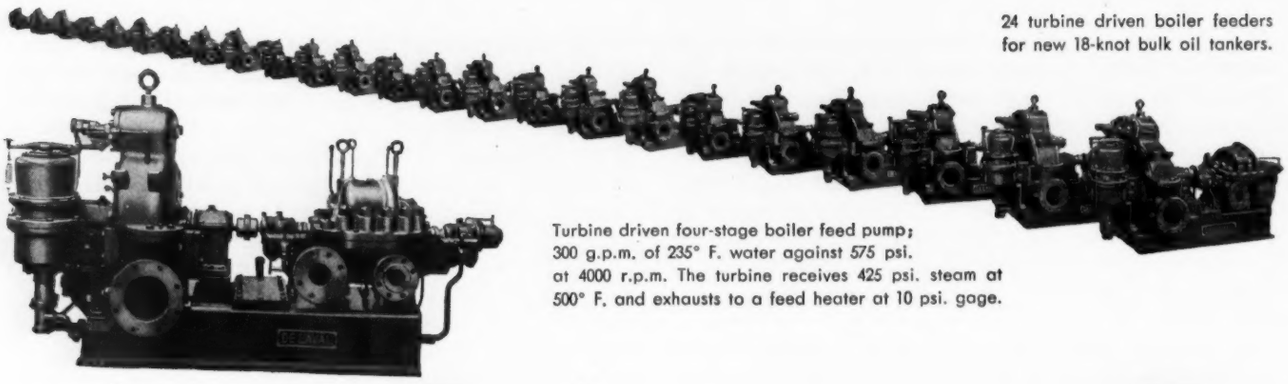
Steam vs. Electric Auxiliary Drives

In regard to auxiliary drives, steam and electric drives each have their advantages and disadvantages. Steam drives have an advantage in ease of speed control but have a higher steam rate than the main generating unit and the economical use of the exhaust steam is often a problem. Mechanical-drive turbines frequently require a speed-reducing gear and this adds to the cost of the unit and is another piece of equipment to be maintained. For a drive requiring a small amount of power it is usually cheaper and simpler to install a motor.

In an industrial plant requiring additional boiler capacity and having a small electric generating capacity, the boiler plant auxiliaries that are electrically driven may increase the demands on the electric generating capacity so that it will also have to be extended. If some of the drives are large, special means may be required to start them without serious disturbances on the power circuits. The hydraulic coupling and the magnetic coupling are convenient for adjustable-speed drives, such as on induced-draft fans when using a constant-speed induction motor. As the power required to drive a fan varies as the cube of the speed, the reduction in efficiency of the coupling as the speed ratio is increased is not important for the small power required at low speed. Both types of couplings, however, add considerably to the cost of the drive.

In selecting the type of drive for auxiliaries, provision for starting the plant from a complete shutdown must not be overlooked. Not considering accidental shutdowns, there are times when it is desirable to shut down an industrial power plant and the auxiliaries should be chosen and arranged so the plant can be readily started. It is awkward to have all electrically driven auxiliaries supplied from a main generator and no steam to operate the generator.

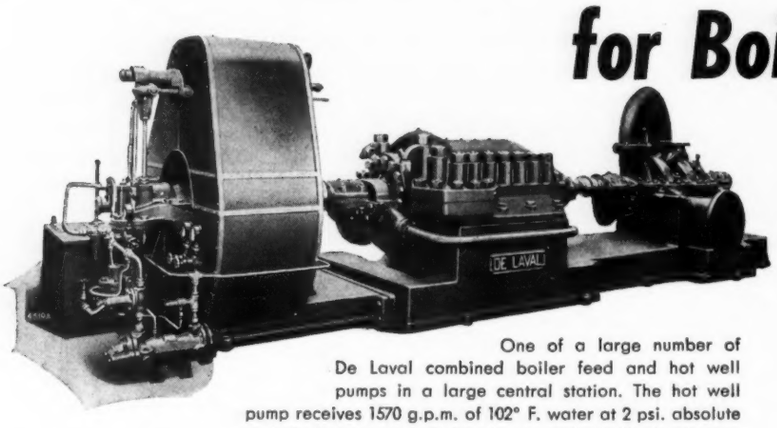
Reliability is probably the most important consideration in the plant and the drives should be selected to maintain continuity of service. It is open to question whether a steam drive taking steam from the mains near the boiler is more dependable than an electric drive properly installed. The additional connections to the piping system increase the probability of trouble, particularly with high pressures but it would seem that there is less chance of sudden shutdown with steam than with electric drive.



24 turbine driven boiler feeders for new 18-knot bulk oil tankers.

Turbine driven four-stage boiler feed pump; 300 g.p.m. of 235° F. water against 575 psi. at 4000 r.p.m. The turbine receives 425 psi. steam at 500° F. and exhausts to a feed heater at 10 psi. gage.

ADJUSTABLE SPEED for Boiler Feed Pumps



One of a large number of De Laval combined boiler feed and hot well pumps in a large central station. The hot well pump receives 1570 g.p.m. of 102° F. water at 2 psi. absolute and discharges against 210 psi. gage. The boiler feed pump receives 1455 g.p.m. of 260° F. water under 169 psi. gage and discharges against 1186 psi. gage. The turbine, running at 1770 r.p.m., receives steam at 815 psi. gage and 900° F. and exhausts against 5 psi. gage.

In feeding a boiler, the point where constant excess pressure is required is at the regulator valve. At maximum load, the drop through piping, feed heaters and economizer may amount to 50 lb. or more, and if pump speed is held constant power will be wasted at lesser loads. Adjustable speed is readily obtained by using a

DE LAVAL TURBINE DRIVE

In large units and where the turbine exhaust is used in feed heaters, turbine drive competes in dollar economy with electric drive, besides offering greater security against shutdown.

With motor drive, variable speed can be had with a D.C. or wound-rotor A.C. motor or by using a hydraulic coupling. Slow speed motors are the most adaptable to speed control, but a De Laval speed increasing gear will permit of selecting the pump speed that is best for efficiency and good construction.

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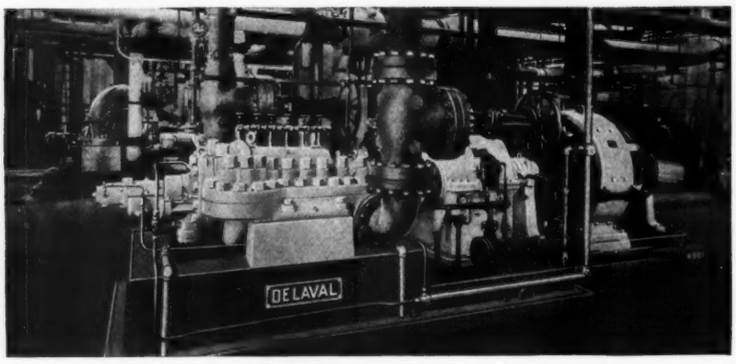

Boiler feed pump driven by variable speed D.C. motor through speed increasing gear. Maximum speed of motor 1200 r.p.m.; pump 2475 r.p.m.; to deliver 1165 g.p.m. of 350° F. water from a suction pressure of 300 psi. to a superposed boiler at 1085 psi. The shaft packings of the pump are protected from high pressure by labyrinth pressure breakdowns and leak-off chambers at each end of the pump.

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MANUFACTURERS OF TURBINES — STEAM, HYDRAULIC, PUMPS — CENTRIFUGAL, PROPELLER, ROTARY DISPLACEMENT, MOTOR-MOUNTED, MIXED-FLOW, CLOGLESS, SELF-PRIMING, CENTRIFUGAL BLOWERS and COMPRESSORS; GEARS — WORM, HELICAL and FLEXIBLE COUPLINGS

Energy and Flow CHARTS for Air Calculations

The accompanying charts, one for computing velocity and the other giving the ratio of flow per unit area to critical flow per unit area as a function of pressure ratio, are based on the more exact but rather involved equation for velocity by de St. Venant and that on quantity of flow. By their use cumbersome calculations may be avoided in determining the velocity and the maximum quantity of air that will escape through a given orifice. Use of the charts, however, is not limited to the type of example given in the text.

By **WINSTON R. NEW**
Research Engineer
Westinghouse Electric & Mfg. Co.

IN CALCULATING energy and flow quantities for gases, the most common of which is, of course, air, engineers frequently find it necessary to employ the more exact equations involving a variable density instead of the familiar $C = \sqrt{2gh}$ and $Q = AC$. These universally recognized relations for incompressible fluids are reasonably accurate for gases only when the velocities remain low, of the order of about 200 ft per sec or less. The more exact steady state equations (1) for velocity, and (2) for quantity of flow, developed in almost every textbook on thermodynamics or fluid mechanics are obtained by combining the momentum theorem (Newton's Second Law of Motion), for a one-dimensional spatial variation of velocity, and equation of continuity with pv^k = constant, expressing the isentropic (frictionless adiabatic) behavior of the fluid.

$$C_2 = \sqrt{\frac{2gk}{k-1} RT_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right] + C_1^2} \qquad (1)$$

$$\frac{A_2 C_2}{v_2} = A_2 \sqrt{\frac{2gk\,p_1^2}{(k-1)RT_1}\left[\left(\frac{p_2}{p_1}\right)^{2/k} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\right] + \left[\frac{C_1 p_1}{RT_1}\left(\frac{p_2}{p_1}\right)^{1/k}\right]^2} \qquad (2)$$

In these expressions the subscripts 2 and 1 designate the final and the initial conditions, respectively; p is the pressure in pounds per square foot, absolute; v represents the specific volume, in cubic feet per pound; T is the absolute temperature in degrees F; A is the area in square feet; C the velocity in feet per second; k is a constant = 1.4; and g, the acceleration of gravity in feet per second per second.

The C_1 term, representing the velocity of approach, is usually considered zero in writing equations (1) and (2) and they then assume a somewhat simpler and more familiar appearance.

The odd powers of the pressure ratios make solution of these equations rather laborious and having been confronted with a great deal of this work, the author has prepared the accompanying charts in order to facilitate making the calculations. On the assumption that other engineers might also find the charts useful in solving various problems they are here reproduced in a size permitting easy and accurate reading.

To employ the velocity chart for any initial temperature other than the 70 F room temperature selected for our calculations, it is necessary only to multiply the chart reading by $\sqrt{\frac{t + 460}{530}}$ where t is the Fahrenheit temperature required.

For the critical pressure ratio ($p_c/p_1 = 0.5283$ for air) the expression $\left[\left(\frac{p_2}{p_1}\right)^{2/k} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\right]$ reaches a maximum, yielding for the critical flow per unit area the well known simple formula (3) where C_1 has been considered zero.

$$\gamma_0 C_0 = \frac{C_0}{v_0} = \frac{3.885 p_1}{\sqrt{RT_1}} \qquad (3)$$

or if the value of R for air be inserted

$$\gamma_0 C_0 = \frac{0.5319 p_1}{\sqrt{T_1}} \qquad (3a)$$

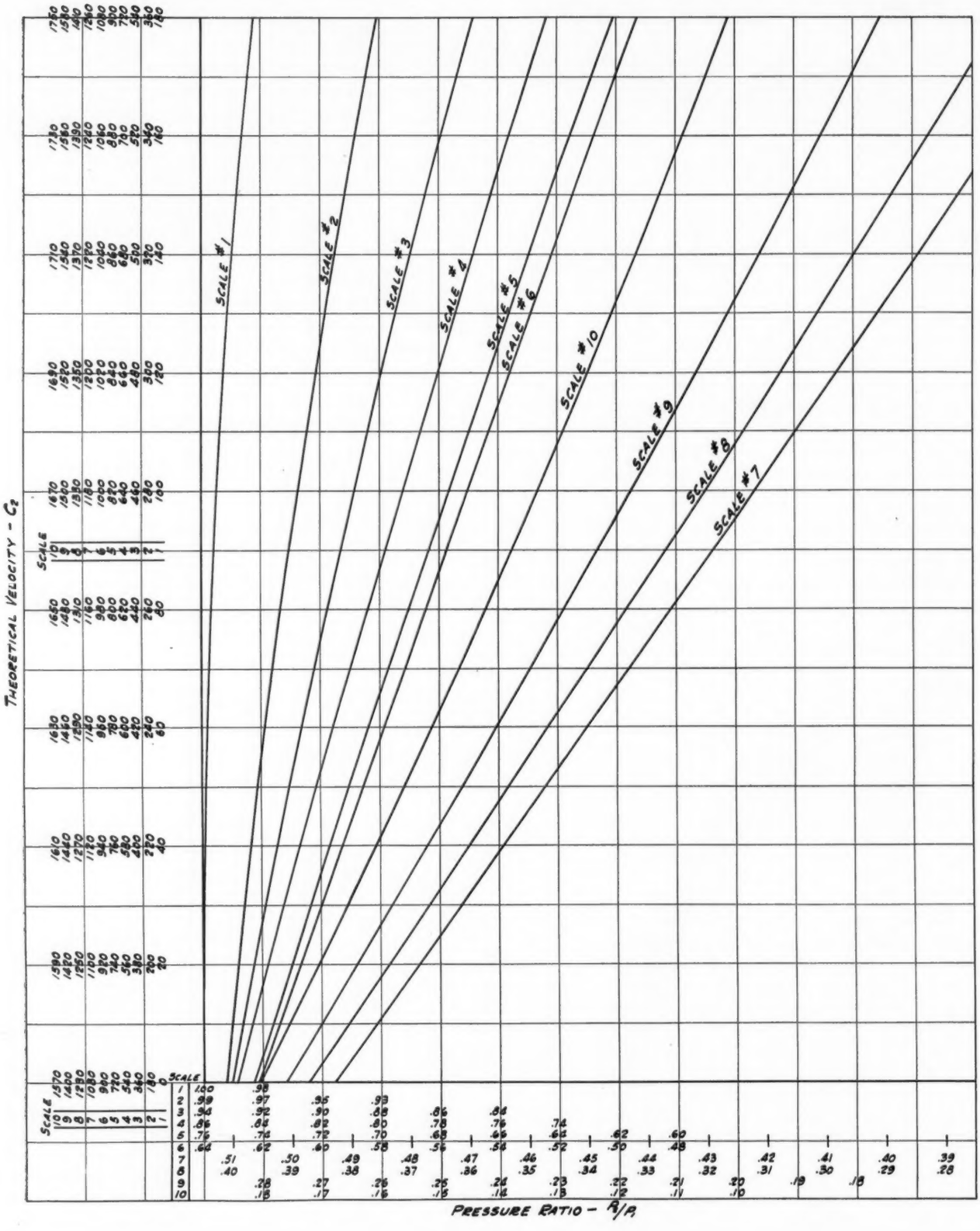
where γ designates density in pounds per cubic foot and is simply $1/v$. The calculation of flow per unit area for any pressure ratio then requires only the product of a figure picked from the chart and the simple formula (3a). A typical example follows:

Example

Suppose it is desired to ascertain the velocity and the maximum quantity of air that could escape to the atmosphere from a one-inch diameter hole in a large reservoir in which a pressure of 10 lb per sq in. gage and temperature of 100 F were maintained.

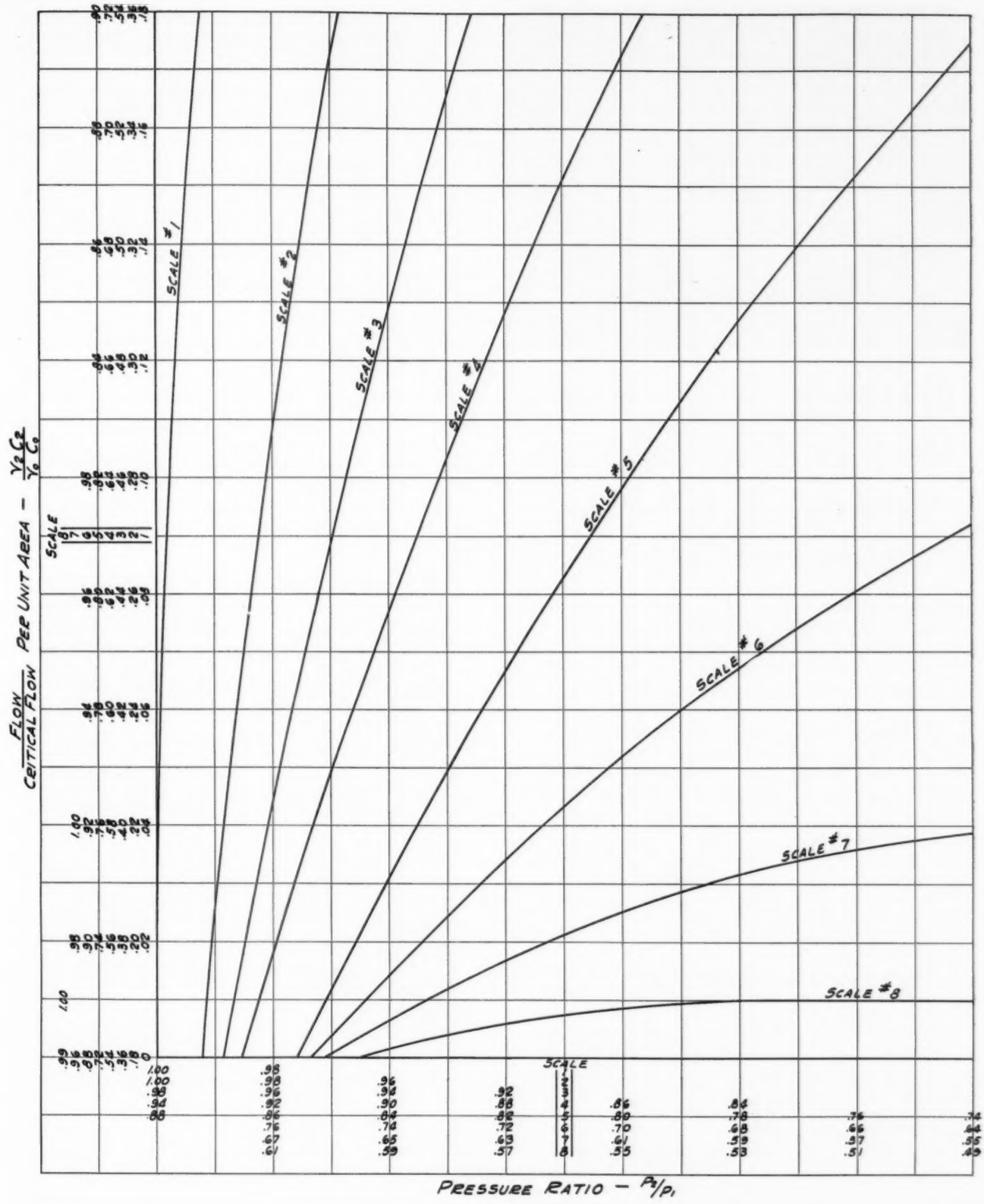
Assuming a barometric pressure of 14.7 lb per sq in., from the velocity chart we find 937 ft per sec corresponding to a pressure ratio $\frac{p_2}{p_1} = \frac{14.7}{24.7} = 0.595$ and an initial temperature t_1 of 70 F. The required velocity is then

$$C_2 = 937 \sqrt{\frac{100 + 460}{70 + 460}} = 963 \text{ ft per sec.}$$



Velocity chart

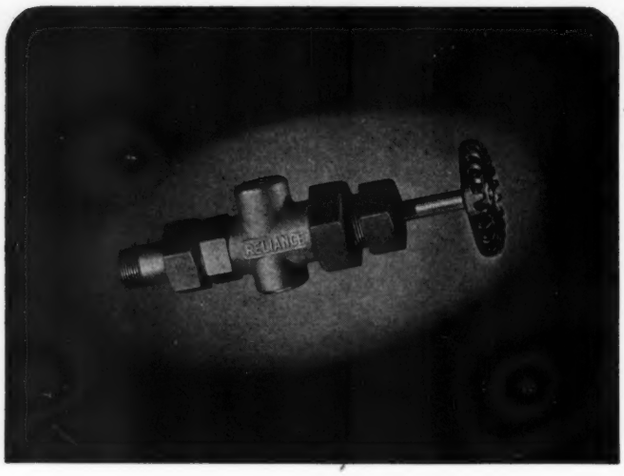
Based on the equation of de St. Venant, $\frac{C_2^2 - C_1^2}{2g} = \frac{k}{k-1} \cdot RT_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]$ solved for $C_1 = 0$, $T_1 = 530$ F abs., $R = 53.35$, $k = 1.4$ and $g = 32.2$



Ratio of flow per unit area to critical flow per unit area as a function of pressure ratio

Based on equation $\frac{\gamma_2 C_2}{\gamma_0 C_0} = \sqrt{\frac{\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}}{\left(\frac{2}{k+1}\right)^{\frac{2}{k-1}} - \left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}}}$

solved for $k = 1.4$



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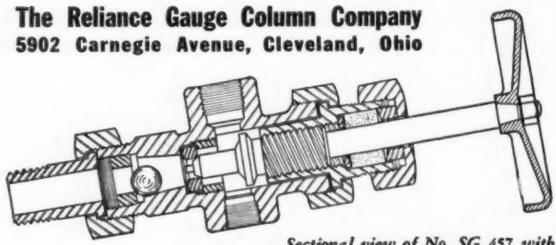

Sectional view of No. SG 457 with ball check cut-off. Metallastic stem packing; union tank connection.

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From the flow per unit area chart we find the ratio 0.99 corresponding to pressure ratio 0.595. The required quantity of flow is then

$$A_2\gamma_2C_2 = \frac{\pi \times 1}{4 \times 144} \times 0.99 \times \frac{0.5319 \times 144 \times 24.7}{\sqrt{560}} =$$

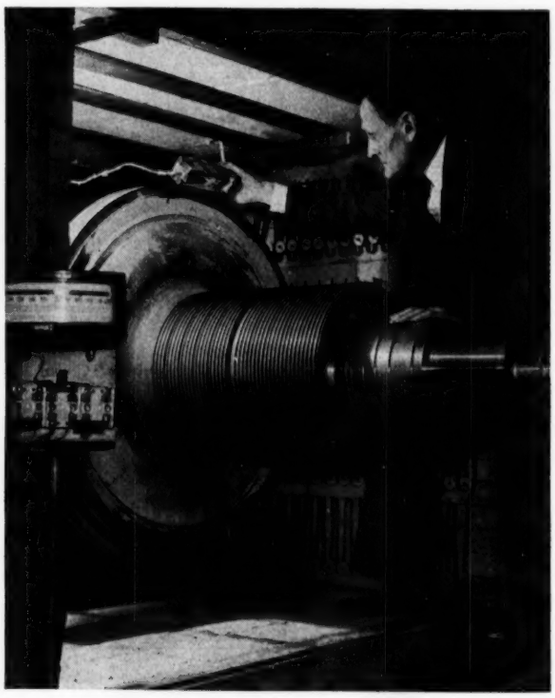
0.432 lb per sec.

To obtain this maximum flow the leading edge of the hole would have to be well rounded in order to avoid the consideration of a contraction coefficient. Friction losses for the example cited would be almost negligible.

The dimensionless "Flow per Unit Area Chart" represents a unique function of k and is therefore applicable, in conjunction with equation (3), to calculations for diatomic gases other than air. It is only necessary to substitute the correct value of R in equation (3) for the diatomic gas considered.

Truing Turbine Shafts

S. Homer Weaver, General Electric engineer who recently received a Charles A. Coffin award for his method of "baking" permanent straightness into steam-turbine shafts and rotors, is here shown inspecting one of the thermocouples before a run.



Shaft in electric oven

In Mr. Weaver's process, the machined-straight shaft or rotor is suspended in a standard lathe bed. The electric oven, sectionalized for different length requirements, is closed about it and controlled heat is applied through strip heaters on the wall. The rotor is turned at 2 rpm and thermocouples riding on it give temperature readings. As the oven heat is raised, the rotor loses its straightness. Sliding rods passed through the wall and riding against the turning rotor indicate increasing shaft deflection as the temperature goes up. Then, at a certain critical temperature, a straightness is restored which is not affected by temperature or temperature changes.

Liability Under Service Contracts

By LEO T. PARKER
Attorney at Law, Cincinnati, Ohio

A review of pertinent court decisions defining the legal rights of seller and purchaser, under contracts concerning plant equipment or sundry appliances.

FREQUENTLY plant owners, engineers and others have legal controversies involving contracts of sale which contain a clause wherein the seller agrees to render service on the equipment for a stipulated period. Generally speaking, contracts of this nature are valid and enforceable by the purchaser, providing he fulfills his own obligations under the contract, but failure of the purchaser to do so places the seller in an advantageous position. If the purchaser breaches the contract the law gives the seller the privilege and opportunity of deciding upon the next step.

Legal Rights of Seller

In such cases the important consideration is: Did either the buyer or the seller fail to fulfill his obligations assumed in the contract? In other words, while a seller is liable for failure to perform service in accordance with the terms of the contract, failure of the purchaser to keep equipment in proper adjustment, if such is required in the contract, also is a breach on the part of the purchaser which renders the latter liable.

For illustration, in *Carisch, 255 N. W. 814*, it was shown that a purchaser bought a machine and paid $500 down. He was to pay the balance in twenty-four monthly installments. In addition to this sum, he was to pay a stipulated amount for service and inspection to be supplied by the seller. The contract stated that the latter was to supply and install the equipment and supervise the necessary repairs and adjustments which, *in his opinion*, were necessary. The contract provided further that the purchaser agreed to keep on hand necessary repair parts and that he would maintain the equipment in good and efficient working order and condition. If a defective condition developed in the machine, he obligated himself to communicate promptly with the seller.

The equipment got out of adjustment and failed to operate satisfactorily. However, the purchaser failed to notify the seller who later entered suit for the amount due.

The court held the seller entitled to recover the entire balance due because the purchaser breached the contract when he failed to notify the former that the machine was out of order and that it could not be repaired immediately.

A purchaser breaches a contract where he fails to make payments on dates specified in the contract; if he fails to accept delivery, as agreed; or if he fails to use the subject of the sale in the manner prescribed in the contract. Under these circumstances the seller may refuse to continue to perform his obligations under the contract and sue the purchaser for damages and profits equal to his financial loss resulting from the breach; he may file suit and compel the purchaser to fulfill the exact terms of the agreement; or, the parties may enter into negotiations and make a supplementary contract.

Rights of Plant Owner

A plant owner who purchases equipment under a contract has certain well-defined legal rights against a seller who breaches any clause in the contract. A review of higher court cases discloses that a purchaser is privileged to cancel a contract and recover damages where the seller fails to make delivery of equipment in the manner specified, or fails in other particular details to fulfill the assumed obligations.

Another important point of law is that although the seller may not give a guarantee, the law implies that the subject of the sale is reasonably fit for the intended purposes. This is applicable to a manufacturer, his agent or a firm that acts as retailer.

Service Contracts

After apparatus is accepted and in operation, the seller is bound by reasonable provisions relating to the service which he has promised to give. Despite this, the courts will not permit a purchaser to practice imposition on the seller, unless an imposition is clearly expressed in the contract. For illustration, in one case a contract of sale contained a clause which "guarantees that the purchaser shall be satisfied."

Although the equipment was reasonably good and worth the purchase price, the purchaser continually, and at unreasonable frequency, requested the seller to supply an unnecessary amount of service in adjusting, repairing and otherwise performing work on it. At last the seller refused to give more service. Then the purchaser notified the seller that he was not satisfied. Since the seller had guaranteed satisfaction the purchaser demanded that the seller take back the equipment and refund the amount paid. He notified the latter that he would not meet other payments due.

The seller filed suit and proved that the appliance had given reasonably good service, notwithstanding the complaints registered by the purchaser. In view of this testimony, the higher court held the purchaser bound to pay the balance due.

On the other hand, if a contract of sale clearly and distinctly specifies the kind of service, on what dates in-

spections, alterations, adjustments and necessary repairs shall be made, then the seller positively is obligated to fulfill the exact terms of this contract.

Used Equipment

While a plant owner who purchases new equipment has an implied guarantee that it is reasonably fit for the intended purposes and worth the purchase price, this rule usually is not applicable to second-hand equipment.

For instance, in *McDonald v. Dorfman, 32 S. W. (2d) 443*, it was disclosed that a seller sold and delivered used supplies, materials and appliances. The purchaser refused payment on the contention that the same were not reasonably worth the purchase price. However, the court held the purchaser liable for full payment and said:

"It is undisputed that the property was second-hand. It was known by the parties to be second-hand and was sold as such. It is undisputed that the purchaser had the opportunity to inspect the property..... When one buys second-hand property, knows it to be second-hand, and either inspects it or has the opportunity to do so, there is no implied warranty."

Law of Fraud

Irrespective of provisions of contracts of sale, or whether the equipment is used, or otherwise, a seller always is responsible for deceit or fraud. For illustration, a rejection of delivery or rescinding of a contract may be based upon fraud on the part of the seller or his agent. However, the legal meaning of "fraud" or "deceit" is widely different from its ordinary dictionary definition. Legally, actionable fraud on the part of a seller means that all of these facts can be proved by the purchaser, as follows:

(1) There must be conduct, acts, language or silence on the part of the seller amounting to a misrepresentation or a concealment of material facts from the buyer. (2) These facts must be known to the seller *at the time and before* the sale contract is completed. (3) That these facts are fraudulent must be *unknown to the purchaser* at the time he made the contract. (4) The defrauding acts of the seller must be done *with the intention, or at least with the expectation*, that the purchaser will be influenced. (5) The conduct or promises of the seller must be relied upon by the purchaser.

Therefore, it is well established that although a seller practices deceit upon the purchaser the latter has no legal recourse if, before the sale was made, he did not believe the deceitful actions or fraudulent statements were true.

Failure to Read Contract

A common source of controversy is where a purchaser endeavors to cancel a contract contending that he was not fully informed of the contents of the contract when he signed it. However, it is well established that a contract is not rescindable, although it is proved that the complaining party was mistaken with respect to the obligations, unless convincing evidence is introduced showing that cancellation of the contract is mutually agreeable to *both* parties. An example of this phase of the law is found in the late case of *Dixon v. Morgan, 285 S. W. 558.*

In this case it was shown that a purchaser signed a contract of sale without taking the necessary precaution to read it carefully. The court held that the general principle of the law is that a contract made through mistake of one party will be canceled *only* when it can be done without doing injustice to either person. Moreover, this rule is applicable although the purchaser testifies that he was deceived by the seller or his agent.

Seller Delays Delivery

A review of recent leading court decisions discloses that a seller may without liability delay delivering under the following circumstances: (1) when the buyer actually consents verbally or in writing to the delay; (2) when the buyer orders a change in the original agreement or specifications which actually delays the delivery; (3) when, after the delay, the purchaser agrees to accept the delivery; (4) when the purchaser performs an act which terminates the seller's obligations; (5) where an obligation mutually agreed upon by the contracting parties, *not* a mere printed notification on the contract form of which the purchaser has no notice, results in the right of the seller to delay shipment; (6) when the purchaser breaches the contract before the date for making delivery arrives; (7) when delayed delivery results from public enemy, as armed forces during a war, or an act of God.

Boiler-Water Problems To Be Discussed at Worcester

Steam blanketing and steam purity, subjects of intense interest to power plant engineers and chemists, are to be discussed under the sponsorship of the Joint Research Committee on Boiler Feedwater Studies at the Spring Meeting of the American Society of Mechanical Engineers. The Boiler Feedwater session will be held on the morning of May 2, and anyone interested, whether or not he is a member of the A.S.M.E., is invited to attend. Non-members will, however, be subject to a nominal registration fee of two dollars.

The discussion on steam blanketing will be based upon a paper by E. P. Partridge and R. E. Hall, of Hall Laboratories, Inc., presented at the 1939 Spring Meeting of the A.S.M.E. in New Orleans, and published in the December, 1939, issue of the *Transactions* of this Society. This paper presented a number of case histories in which the failure of boiler tubes was attributed to the development of a more or less continuous blanket of steam. It is expected that additional cases will be discussed at Worcester, together with the steps taken to minimize this type of failure in several plants.

Steam purity will be the subject of a paper by M. C. Schwartz of Louisiana State University, well-known as the originator of the Schwartz and Gurney procedure for the determination of very small amounts of dissolved oxygen in boiler feedwater. The demand made upon modern boilers to produce steam free from even a mist of mechanically entrained boiler water has directed attention in recent years along three main lines, the chemical control of boiler-water conditions to minimize foaming, the development of mechanical devices for separating suspended droplets of boiler water from steam, and the improvement of methods of testing for dissolved substances in steam condensate. Discussion along each of these lines is anticipated at the Worcester Meeting.

Low-Temperature Carbonization

In a paper at the February meeting of the American Institute of Mining and Metallurgical Engineers in New York, C. E. Lesher reviewed the commercial development, over a six-year period, of the production of low-temperature coke by the Disco process. This process operates under the Wisner patents, with improvements and changes made by the Pittsburgh Coal Carbonization Co. Briefly stated, it consists of the continuous heating and carbonizing of undersize coal in an inclined revolving retort to form low-temperature coke in rounded, homogeneous ball-shaped pieces varying from 1 to 8 in. in diameter. Mr. Lesher offers the following explanation of what takes place in the formation of these coke-balls:

How Coke Balls Are Formed

"The mixture of preheated, finely divided coal and breeze is conveyed by the revolving action of the inclined retort toward the discharge end. At the point where the incoming material meets the steel the temperature of the shell is about 800 F and that of the coal mixture is increased as it progresses until it attains the temperature at which it becomes soft or plastic. Yellowish hydrocarbon vapor is given off as the coal softens. In the dry state the feed does not stick to the steel. The hot, dry, fine coal and breeze flow freely. At above 700 F the coal softens, gives off hydrocarbon vapors and distillation has begun; in fact, some hydrocarbon vapors are given off below 600 F.

"As the coal softens it becomes sticky. The whole mass does not melt into a fluid state, but some constituents of the coal soften, wet the surfaces of adjacent or inert particles and bind the whole into a loosely coherent mass. In this condition its coefficient of friction on steel is greatly increased. Instead of a thin layer, loosely flowing, there is a thickening of the bed, because the increase in friction not only retards its flow down the retort but increases the length of climb up the arc of the revolving shell.

"Revolving action of the shell raises the mass and as it falls over it is kneaded and disintegrated into smaller masses, quite irregular in shape and size. The segregation into small masses that later become hardened as coke balls is not dependent on so-called "snowballing" action; that is, the larger masses are not the result of small pieces rolling down a surface of material and increasing in size. The size of the masses that finally merge as coke balls is determined by the agglutinating property of the coal at the time it reaches its softening temperature in the retort. The structure of the low-temperature coke is determined by the character and proportion of inerts mixed with the coal and by the rate and maximum temperature of heating; also, the effective control of the rates and temperatures in the preplastic, plastic and postplastic stages. The volatile content of the product depends on the treatment in the retort. It has been found that a product with satisfactory structure will contain about 16 per cent volatile."

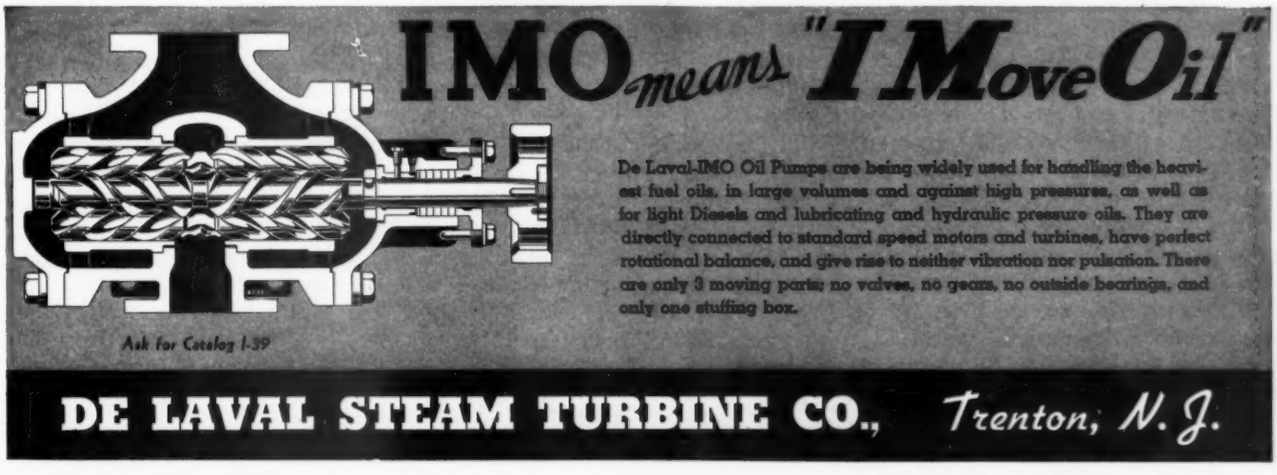
By-Products from Process

The crude products obtained by this process in addition to coke are tar, light oil, liquor (aqueous liquid) and non-condensable gas. Thus far, tar and coke are the only products sold commercially. The gas, containing the light oil vapor, is used in heating the retorts at the Champion Plant and the excess is wasted to the atmosphere. The following table shows the yield per ton of coal as received.

YIELD OF PRODUCTS FROM 2000 LB OF COAL AS RECEIVED

Product	Per Ton Coal
Coke	1440 lb
Gas	3720 cu ft
Tar	15.8 gal
Light oil	2.5 gal
Liquor	6.6 gal

The process is continuous and the capacity per unit is 160 tons of coal per day. Generally speaking, the coals that are most suitable for this process are fines that in recent years have been marketed at less than a dollar per ton. The market value of Disco is now $4.50 f.o.b. ovens in the East but will average over a year between $3.25 and $4.00 per ton. The margin between coal cost and sales realization of the coke exceeds $2.00 per ton of product. The value of the tar from producing a ton of Disco averages 75 cents. Thus there is $2.75 to cover operation, fixed charges and profit.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Steam Formation in a Vertical Boiler Tube

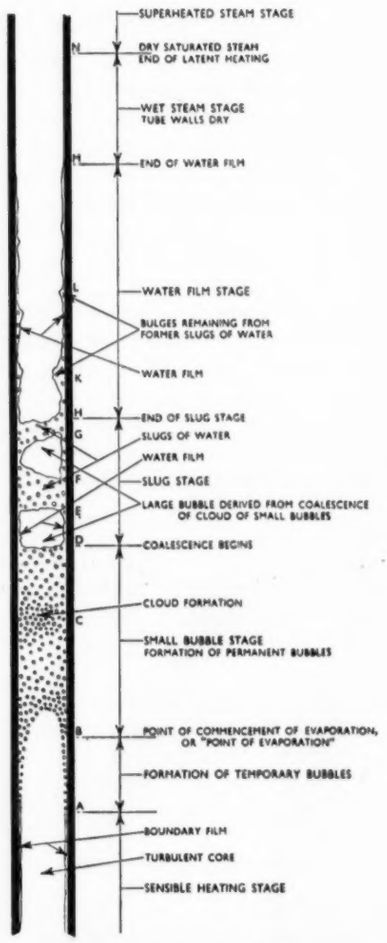
In a paper before The Institution of Mechanical Engineers (Great Britain) on March 1, 1940, W. Y. Lewis and S. A. Robertson presented an extensive analysis of the circulation of water and steam in water-tube boilers. The paper covered 31 printed pages, including numerous charts and mathematical expressions, rendering any comprehensive abstract impracticable. However, the following explanation of the mechanism of the change from water to steam in a vertical tube is of particular interest. This was prepared by the authors to illustrate the findings of such investigators as Barbet (1914), Badger (1937), Boarts, Badger and Meisenburg (1937) and Stroebe, Baker and Badger (1939), as well as the results of their own experiments.

Up to Section *A* there is water only in the tube. At the usual velocities the flow will be turbulent. On any cross-section there is a thin boundary layer, next to the tube wall, moving in a laminar or viscous flow, then a transition zone, then the turbulent core. The temperature drop, between the inner tube surface and the water core, is concentrated almost entirely in the boundary layer, across which heat moves mainly by conduction. In the turbulent core, heat is quickly distributed by eddy currents, so the temperature is approximately uniform. Above *A*, when the temperature of the water immediately in contact with the tube is high enough, small bubbles of steam form on the tube wall.

Due to the surface tension of the water enclosing a spherical bubble of steam the pressure in the interior of the bubble is greater than the pressure in the water, the excess being inversely proportional to the radius of the bubble. If no particles of air or other nuclei are present to start the formation of bubbles, boiling will not begin until the temperature is raised much above the point corresponding to the external pressure; that is, the liquid becomes superheated, above saturation temperature. Once formed, the bubbles would be highly unstable, for, as the radius increases, the tension in the envelope becomes less and less able to balance the excess of pressure within it, and the expansion into vapor would take place with explosive violence.

A bubble formed on the heating surface will grow until it is swept off by the moving liquid film, or until it bursts through the film. The bubbles, once dislodged, travel inward toward the core of the tube. A chain of bubbles may originate from any particular nucleus. At first, this core is not sufficiently hot to enable the bubbles to exist, and they collapse after a brief travel. This occurs at Section *AB*. By condensing and giving up their latent heat, the bubbles contribute toward raising the core temperature to saturation temperature.

Above *B*, the core temperature is sufficiently high to enable the bubbles to continue to exist and they increase in both number and size. They rise and their upward velocity is greater than that of the water. Although in the earlier stage the distribution of bubbles is fairly uniform, after a point they seem to gather in clouds, as at *C*, separated by sections in which they are less numerous.



Change from water to steam in a vertical tube
The vertical scale is smaller than the horizontal scale

Various observers have described how, after a certain point *D*, the bubbles of each cloud coalesce to form a single large bubble. The water displaced during this action forms a slug above the steam bubble, which is pushed along by the latter. Between *D* and *H* there is a section of slug action, the tube being occupied by alternate slugs of water and steam traveling with equal velocity. However, there is still a film of water on the tube wall. The water slugs become thinner as they travel upward, due to evaporation, until they finally break up at *K*.

From *H* to *M* there remains only a film of water on the

tube wall, with occasional bulges, as at K and L. This water film travels upward but at less speed than the steam speed. It evaporates by nucleate boiling, the small bubbles of steam forming on the tube wall and being either swept off by the movement of the film or bursting through the film. During the breaking up of the slugs and during boiling of the film on the tube wall, the bursting of steam bubbles into the general column of steam entrains particles of water into the core. These travel upward with the steam and form one of the principal elements in the wetness of the steam. Above M there is steam containing drops of water passing up the tube. These gradually evaporate until the steam becomes dry saturated at N. Above this, superheating begins.

Commenting on high-head boilers, the authors point out that the purpose is to increase the length of tube containing steam of maximum dryness and minimum density, counterbalanced by an equal length of downcomer tube containing water. The increase in head is intended to increase the velocity of flow in the riser tube and hence improve the circulation. However, it is their contention that the velocity in the tubes of such a boiler is not more than 25 per cent greater than in a standard U-tube type, as much of the extra head is required to overcome friction in the added length of tubing.

Combustion of Pulverized Coal

In *Die Wärme* for November 4, 1939, Dipl.-Ing. Jos. Ehmig gives a picture of some of the reactions which occur when pulverized coal is being burned and points out the desirability as well as ways and means for speeding up the combustion process. This is a chemical reaction taking place primarily on the surface and which can be accelerated greatly by preheating fuel and primary air to a high temperature, by increasing the surface of pulverized coal, that is, increasing the fineness, and by a high secondary air temperature.

By means of the following formula developed by Rosin, the time of combustion can be calculated for a furnace temperature of 2372 F and for lignite:

$$Z = \frac{133 \times 10^8}{(F/G)^{1.8}}$$

where Z = time for combustion in seconds
F = surface in inch²
G = weight of coal particle in lb

As soon as the moisture in the coal is driven off, the temperature rises rapidly and at the same time decomposition of the complicated structure takes place.

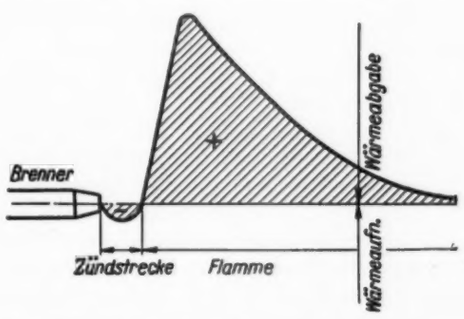


Fig. 1—Heat flow of combustion speed diagram
(Brenner) Burner; (Zündstrecke) ignition period; (Flamme) flame; (Wärmeaufn) heat absorption; (Wärmeabgabe) heat release

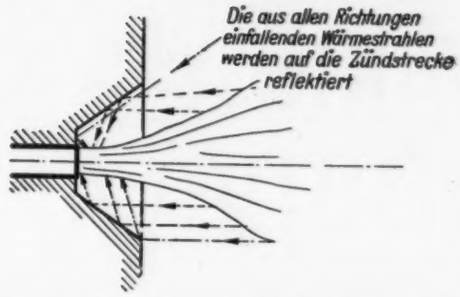


Fig. 2—Arrangement of burner muffle, depending on burner design, form of flame and pulverized coal characteristics
(Die aus allen Richtungen einfallenden Wärmestrahlen werden auf die Zündstrecke reflektiert). Heat rays coming from all directions are reflected on the cold section of the coal stream

It has been found that the higher the ignition temperature, the more symmetrical and simple the chemical structure of the substance. The moisture absorbs heat during the ignition period (see Fig. 1) and it is desirable to keep this period as short as possible by drying of the coal, by limiting the amount of primary air to that required for the combustion of the volatile and by employing a refractory muffle (see Fig. 2).

High ash content lengthens the combustion process and highly preheated air must be used.

The forming of a CO_2 envelope around the coal particle prevents further contact with O_2; hence the importance of turbulence in the flame. Also the high partial pressure of CO_2 already present must be overcome when new CO_2 is formed. Increasing excess air as well as introducing steam will lower the partial pressure of the CO_2. Steam also acts as a catalyst.

To facilitate complete combustion, the larger coal particles must be kept in suspension for a longer period. This can be accomplished by introducing the secondary air at the proper location.

The mean time of combustion which is at the disposal of the pulverized coal particle can be calculated from the mean length of time which the gases take to pass through the furnace.

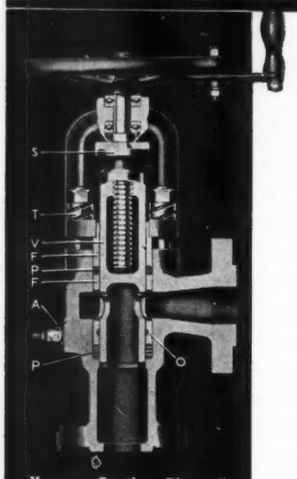
$$Z = \frac{3600V}{BG_1(0.935 + 0.00204t_b)n} \text{ or } Z = \frac{3600(H_u + L_h t_h C_p + L_F t_F C_p')}{bG_1(0.935 + 0.00204t_b)n}$$

where Z = Mean burning time of the coal particle in seconds
H_u = Lower heat value in Btu per pound
V = Effective furnace volume in cubic feet
B = Pounds of fuel as fired per hour
G_1 = Volume of wet products of combustion cu ft per lb at 32 F
t_b = Mean furnace temperature in deg. F
t_h = Preheated air temperature in deg. F, above 32F
L_h = Volume of preheated air in cu ft per lb fuel at 32 F
L_F = Volume of leakage air in cu ft per lb fuel at 32 F
C_p = Mean specific heat of preheated air between 32 F and t_h, Btu per cu ft per deg F
C_p' = Mean specific heat of leakage air between 32 F and t_F, Btu per cu ft per deg F
b = Combustion rate in Btu per cu ft, furnace volume per hour
n = Pounds of coal burned per pound of coal fired (for 5 per cent carbon loss n = 0.95)
t_F = Temperature of leakage air deg F above 32 F

Valuable conclusions can be drawn from these formulas for the design of furnaces. Tendency must be in the direction of creating furnace temperatures as high as possible as this presents the least expensive way to shorten combustion time, which in turn makes for shorter flame lengths and therefore smaller and cheaper furnaces. Furthermore, the question exists whether or not electro-

YARWAY EQUIPPED . .

A Mark of Good Engineering



Yarway Seatless Blow-off Valve. Operation: After Valve is closed, shoulder S on plunger V contacts with upper follower gland F, forcing it down into body and compressing packing P above and below port. Annular groove O connects with Alemite fitting A for lubricating plunger and packing. Yoke springs T maintain continuous pressure through follower gland F on packing rings P.

Yarway Blow-off Valves are used singly or in tandem in more than 12,000 plants in 67 different industries . . . Regarded as a standard of quality by leading steam plant designers and builders of steam generating equipment...Selected for Federal, State and Municipal Institutions...Built for all pressures up to 2,500 lb. . . . Write for Catalog — Section B-420, up to 400 lb. pressure; Section B-430 for higher pressures.

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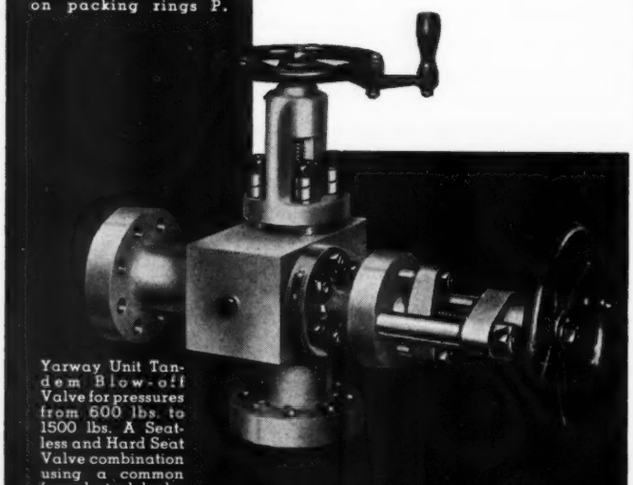

Yarway Unit Tandem Blow-off Valve for pressures from 600 lbs. to 1500 lbs. A Seatless and Hard Seat Valve combination using a common forged steel body.

YARWAY

BLOW-OFF VALVES

dynamic forces could not be utilized which by more intimate mixing would accelerate combustion.

It is further important, especially for low volatile coal, that, with water-cooled furnaces, some portions of the wall remain refractory covered to act as ignition surface at low loads.

Another possibility to increase rapidity of combustion is to keep the furnace under higher pressure. This has been done in the Velox boiler which carries a pressure in the combustion chamber. This accounts in part for the high heat release of 340,000 to 500,000 Btu per cu ft in the Velox boiler furnace.

Acceptance Tests for Steam Boilers

A review of the new British Standards Institution Code for commercial acceptance tests of steam boilers is given in the March issue of *Engineering and Boiler House Review*. The primary object of the Code is to indicate the methods to be employed and the data to be obtained when carrying out a simple efficiency test at minimum cost, but it is not intended to be applied to comprehensive tests on large central-station boilers where a technical staff and extensive equipment are available for testing. Briefly, some of the requirements are as follows:

WEIGHING FUEL—The fuel is to be weighed on calibrated portable platform scales having a maximum capacity of 400 to 500 lb, the coal to be handled in wheelbarrows or boxes, the tare weight of which is to be checked at the end of each hour. Where riddlings are refired their introduction should be distributed over the test run so as to insure uniform combustion conditions.

ANALYSIS OF COAL—Unless the coal is known to have special characteristics, it is not necessary to incur the expense of having an ultimate analysis made. To obtain the hydrogen and carbon content reference can be made to data compiled by the British Standards Institution.

FEEDWATER—This is to be measured by weighing the water in tanks on scales or by means of calibrated tanks with periodic corrections for temperature. As an alternative, the water may be measured by a calibrated venturi meter or orifice, with the manometer readings taken every 30 sec and each ten readings averaged and corrected for temperature. Care is to be taken that the venturi meter or orifice is located in an uninterrupted straight length of feed line. Moreover, this means of metering is to be employed only where the feedwater is supplied by centrifugal pumps. If an integrating type of meter be used it must be calibrated immediately before and after the test run.

TEMPERATURE MEASUREMENTS—For steam and water, and in some cases gas and air at relatively low temperatures, glass thermometers are recommended, and for gas and air at high temperatures thermocouples or electric-resistance thermometers are prescribed. In the case of mercury thermometers, where the temperatures to be measured do not exceed 650 F, the well is to be filled with mercury; above this, solder is to be used, care being taken to remove the thermometer before taking the boiler off the line to avoid solidification of the solder and consequent breakage of the thermometer. Steam cylinder oil may be used up to 600 F. The reason for not permitting a mercury-filled well for temperatures above 650 F is that at 675 F poison vapor is given off.

WEIGHING ASHES—Ashpits or clinker hoppers are to be emptied 15 min before the start and the end of a run,

and the ash weighed dry, after which it is crushed and a representative sample taken for determining the combustible content. If it becomes necessary to quench the ash before weighing, the sample is to be taken while weighing and sealed in a container so that no moisture is lost.

DURATION OF TEST—Where the unit is fired by a chain- or traveling-grate stoker the test is to be continued until the fuel burned per square foot of grate is equivalent to not less than 25 lb for every inch of fuel bed thickness, with a minimum of 4 hr. Thus with a 6-in. fuel bed, the test would continue until 150 lb of coal had been consumed per square foot of grate, and at a burning rate of 30 lb per sq ft per hr, the test would last 5 hr.

With pulverized-coal and direct-fired units, 4 hr, with 2 control hours is stated to be sufficient, although a somewhat longer period is prescribed for units employing the storage system. The Code suggests that, if possible, the load should be held near that of conditions for 24 hr preceding the test. When this is impossible, such conditions should be maintained for an hour before and after the test.

EFFICIENCY—It is observed that even with the best possible care in making such a test, as prescribed by this Code, it is unlikely that an accuracy closer than plus or minus 2 per cent will be attained, and that the allowable tolerance should be agreed upon in advance by the parties concerned.

Creep at High Temperatures

In a recent paper before the North-East Coast Institution of Engineers and Shipbuilders, reproduced in *Engineering* (London) of March 15, H. J. Tapsell reviewed the creep of various steels at elevated temperatures, including the results of some late investigations. The following are excerpts from Mr. Tapsell's paper:

In recent years, research has been made into the possibilities of improving carbon steels by the addition of small quantities of alloying elements, and molybdenum appears to be the most effective single addition. Molybdenum steel containing about 0.5 per cent to 0.8 per cent molybdenum is finding use in steam power plants operating at 900 F and above. Its creep strength exceeds that of carbon steel and, in other respects, it has much the same properties as carbon steel. A comparison of four wrought molybdenum steels, each containing about 0.5 per cent molybdenum, on the basis of the creep rate at 500 hr for a stress of 9 tons per sq in. at 550 C (1022 F) is given in the table. All the steels were made in high-frequency electric furnaces and were tested in the normalized condition.

COMPARATIVE CREEP RATES OF FOUR MOLYBDENUM STEELS

Steel		Creep Rate at 500 Hr Stress: 9 Tons per Sq In. Temperature 550 Deg C
Carbon Per Cent	Molybdenum Per Cent	In. per In. per Hr
0.11	0.52	6.8×10^{-4}
0.185	0.50	1.1×10^{-4}
0.16	0.62	0.4×10^{-4}
0.40	0.55	7.3×10^{-4}

Creep tests on the molybdenum steels at stresses other than 9 tons per sq in. have been made and show the same sort of variation among the four steels as recorded in the table, but the examination of a larger number of steels would be necessary to determine whether or not the variations shown are representative. However, from these and other tests, it appears that low carbon is advantageous in both 0.5 per cent and 1 per cent molyb-

always takes clear thinking, courage and imagination. And it takes something else—a sufficient knowledge of the possible leaks through which the invisible dollars might creep out.

In any existing steam plant, ultimate economy of operation depends upon so many unseen factors in the fuel markets and in the plant itself, that many people are lulled into thinking that just because a plant operates smoothly and at no higher cost than last year, it must be economical.

Repeated experience has shown that an outside professional organization, like the staff of the Fuel Engineering Company, trained to find the invisible dollars through thirty-two years of experience in analyzing plant performance and in guiding fuel selection, *does* find opportunities for economy, which have escaped attention.

One of the products of long and varied experience is the ability to quickly cut through the maze of non-essentials, to recognize those details that are in order, and get at the previously unseen leaks, without wasting one's own time and the client's money.

It is this faculty, won by long training, that has brought to us the outstanding clientele, and the continued patronage of leading industrial corporations which we enjoy year after year.

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denum steels, and that 0.8 per cent or 1 per cent molybdenum gives somewhat better results than 0.5 per cent.

The addition of a small percentage of vanadium to molybdenum steel appears to produce further improvement in creep properties. Molybdenum-vanadium steels of different composition are now being studied by the author. Some chromium-molybdenum steels have strength at high temperatures similar to the plain molybdenum steels and have the advantage of better resistance to oxidation and corrosion. Steels containing about 6 per cent chromium with about 0.6 per cent molybdenum have very good resistance to corrosion, but have less resistance to creep than those in which the chromium is 0.8 to 1 per cent. In the hardened and tempered condition, molybdenum-containing steels are used as bolt materials for high-temperature bolted connections.

At moderate superheat temperatures, cast carbon and molybdenum steels are inferior to the wrought steels, but this inferiority decreases as the temperature increases, and there is little to choose between the two states at about 1000 F. Grain size has an influence on the creep properties of a steel, and it has generally been supposed that large grain size is an advantage. A 0.4 per cent carbon steel coarsened by heating at 1100 C before normalizing showed an improvement in creep properties at 450 C (842 F) as compared with its properties in the normalized condition, but another such steel showed the reverse effect. The quantitative influence of grain size is, however, difficult to assess, since marked structural differences often exist coincident with differences in grain size.

Power Consumption of Boiler Feed Pumps

In a paper before the Association of Large Boiler Users in Germany, reproduced in the December issue of *Archiv für Wärmewirtschaft und Dampfkesselwesen*, W. Allrich discussed the power requirements of boiler feed pumps in relation to the manometric delivery head. For installations of 425 to 570 lb steam pressure these power requirements amount to about 0.75 kwhr per 1000 lb of steam, and for high-pressure installations having natural circulation the power requirements average about 2.25 kwhr per 1000 lb of steam. Those for forced-circulation boilers are about 0.25 kwhr per 1000 lb of steam greater than the natural-circulation type because of the increased resistance of the circuit.

In determining these values the efficiency of the motor was taken at 95 per cent, that of the pump 75.5 per cent for pressures of 570 lb and 67.5 per cent for pressures over 1600 lb. The latest designs of high-pressure boiler feed pumps have slightly higher efficiencies.

Pumps with smaller deliveries (about 100,000 lb per hr) and final pressures between 570 and 1450 lb, with speeds up to 6000 rpm, show efficiencies from 2 to 6 per cent higher than those having speeds of around 3000 rpm. Those with deliveries of around 500,000 lb per hr and final pressures of over 1600 lb and a speed of 5000 rpm have an efficiency about 2 per cent higher than pumps running at 3000 rpm since the runner diameter is considerably smaller.

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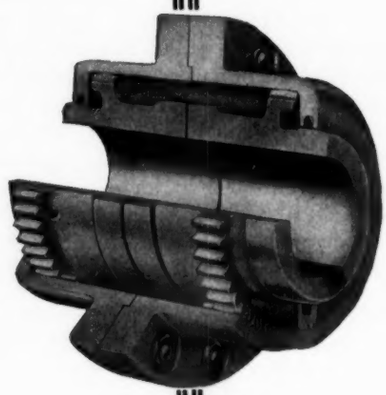

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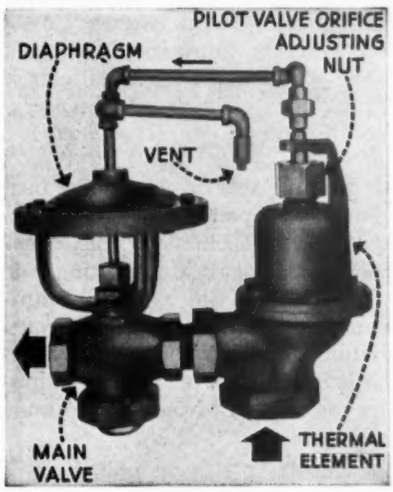
NEW EQUIPMENT

Air Eliminator

A new automatic valve, described as an Air Eliminator, is announced by Armstrong Machine Works. Typical industrial uses include removal of air from steam-heated retorts, paper machines, creosoting cylinders and on-and-off steam mains. Operation is automatic and is based on the fact that air in a steam system has the effect of lowering the steam pressure and hence its temperature. A mixture of air and steam entering the 2-in. inlet cools the thermal unit, which contracts and opens a needle valve. This, in turn, applies pressure to the diaphragm which is depressed, thus opening the main valve and permitting the steam-air mixture to escape through a $1^1/_4$-in. outlet. When the air has been exhausted, steam alone envelopes the thermal element, causing it to expand and close the needle valve. With pressure relieved above the diaphragm, internal pressure forces the



main valve to close. To make sure that this valve remains closed when the steam is turned off, a small compression spring is placed under the end of the valve stem.

Duplex Recorder

A new duplex recorder for recording simultaneously kilowatts and reactive kva or kilowatts and frequency, or volts and amperes is announced by the General Electric Company.

The recorder consists essentially of two separate Type CD instruments placed side by side in one case and arranged to record on a single chart. Timed by a high-torque telechron motor, or an 8-day spring clock, the chart carriage feeds the paper at 1, 2 or 3 in. per hr. Metal pins on each end of the timing drum, engaging in holes on each side of the record paper, assure a positive drive. Each element, equipped with a conventional pen and inkwell, is calibrated for a 4-in. ruled section of the 10-in.-wide record roll. Instead of a single wide chart, a separate standard Type CD chart carriage and chart may be had for each element.

It is expected to have a wide use wherever it is desirable to obtain synchronism between two separate records, or where space economy makes it necessary to place two separate recording mechanisms in a single case.

Fan for Heating, Ventilating and Industrial Use

Reduced power consumption, quiet operation and reduced maintenance are claimed for "Silentvane" Design 7, the

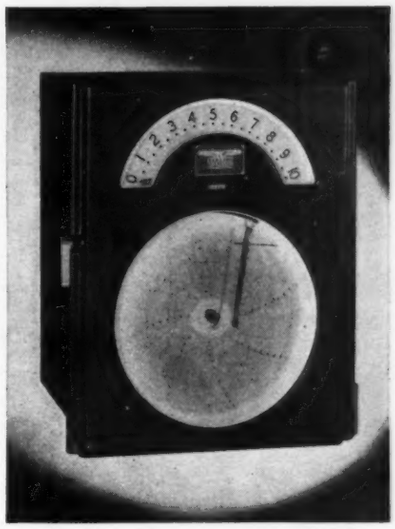
latest addition to the line of B. F. Sturtevant centrifugal fans. Available in all standard discharges, single and double width, single and double inlet, with wheel diameters ranging from $14^5/_8$ in. to 87 in. and static pressures up to 16 in. wg, the new fan is designed for slow rotative and peripheral speeds. It has self-limiting horsepower characteristics with gradually increasing horsepower and volume until the point of maximum efficiency is reached; then decreasing horsepower for larger volumes. Mechanical efficiencies in excess of 70 per cent are obtained over 53 per cent of the operating range, 75 per cent over 41 per cent range and 80 per cent over 21 per cent of the range.

Linameter

The Cochrane Corporation announces a new type of meter, the "Linameter," which is designed for the measurement of fluids having characteristics of viscosity, corrosiveness and solubility such as fuel oil, ammonia, hot tar, etc., which are beyond the scope of the conventional orifice type flow meter.

This new meter is of the area type, with meter body installed as an integral part of the pipe line and containing a weighted disk positioned by the velocity of fluid through a tapered throat section in such a manner that the disk travel is directly proportional to flow rate.

Among the features which distinguish the Linameter are omission of U-tubes, mercury and pressure connecting lines.



Mechanical Ring-Balance Meter

A new mechanical meter of the ring-balance type for metering steam, water, gas, air, etc., at static pressures up to 1000 lb per sq in., has been announced by Republic Flow Meters Co. This new meter will give full-scale readings on differentials as low as 3 in. of water, which is ideal for low-pressure gas and air measurement. The meter is housed in a rugged steel case suitable for exposed locations. It is easily adjusted by changing a calibrating weight, has a 12-in. evenly graduated chart and is furnished with any combination of indicator, recorder and cyclometer type integrator.

Phosphate Comparator

A new junior phosphate comparator set has recently been brought out by W. H. and L. D. Betz. This simplified instrument is an adaptation of the colorimetric method of determining soluble phosphate and it is intended for use in plants where time or personnel limitations make more elaborate methods impractical. By its use complete determination of the soluble phosphate content of boiler water can be accomplished within 3 to 5 min. It is regularly furnished with two standards, 30 to 60 ppm, permitting control of the phosphate within that range. Standards of 20 and 40 or other specified ranges can be supplied if desired.

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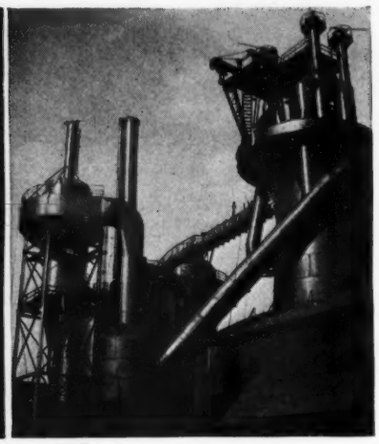

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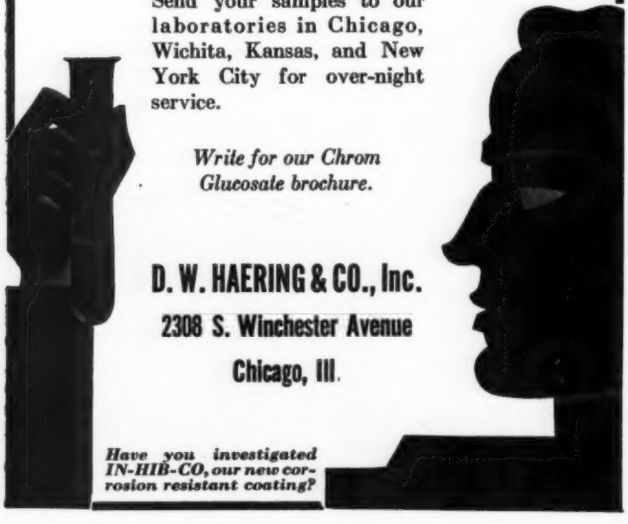

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Have you investigated IN-HIB-CO, our new corrosion resistant coating?

Personals

Asa Binns, formerly chief engineer of the Port of London Authority and now a consulting engineer, has been elected president of the Institution of Mechanical Engineers (Great Britain) for the year 1940.

M. D. Church, president of Moore Steam Turbine Corporation, Wellsville, N. Y., has been elected a vice president of Worthington Pump and Machinery Corporation, of which the former company is a subsidiary.

Harold F. Hagen, vice president and director of research of the B. F. Sturtevant Company, was one of the American scientists recently honored as a pioneer fan engineer by the National Association of Manufacturers on the occasion of the 150th anniversary of the U. S. Patent office.

A. W. Thorson, for several years past with The Detroit Edison Company as fuel engineer, has lately left that company to join the Chesapeake and Ohio Railway Company as assistant fuel service engineer. He is located at Detroit.

James H. Polhemus, for the past four years executive vice president of the Portland Electric Power Company and its affiliates, the Portland General Electric Company and the Portland Traction Company, has been elected president of the Portland General Electric Company to succeed Franklin T. Griffith, who becomes chairman of the board of directors.

Prof. John I. Yellott, who for several years past has been teaching steam power, thermodynamics and heat transfer at Stevens Institute of Technology, has been appointed Director of Mechanical Engineering at Armour Institute of Technology, Chicago. The appointment will become effective in September, concurrent with the beginning of operation of the new Illinois Institute of Technology—the new engineering school brought about by the Merger of the Armour and the Lewis Institutes.

Obituaries

Walter F. Keenan, Jr., vice president in charge of engineering of Foster Wheeler Corporation, died suddenly at his home in Pelham Manor, N. Y., on March 18 at the age of 54. A graduate of the University of Pennsylvania in 1908, he had been associated with Foster Wheeler Corporation, and its predecessor, the Power Specialty Company, for 29 years. He had long been a member of the American Society of Mechanical Engineers and at the time of his death was president of the American Boiler Manufacturers Association and Affiliated Industries.

Ivan E. Powell, supervisor of steam generation for the Rochester Gas & Electric Corporation, died from a heart attack at Norlina, N. C., on March 24 while motoring to his home from St. Petersburg, Fla. Mr. Powell was a graduate of Mt. Union College and held an electrical engineering degree from Ohio State University. With the exception of three years with the

Westinghouse Electric & Mfg. Co. shortly after leaving college, his engineering career was spent with the Rochester utility. He was a member of the American Society of Mechanical Engineers, National Association of Power Engineers, Universal Craftsman Council of Engineers and the Rochester Engineering Society.

Business Notes

Underground Steam Construction Co., Boston, Mass., subsidiary of E. B. Badger & Sons Co., announces the appointment of Charles E. Mayette to its sales engineering staff. Mr. Mayette's activities will be mainly in connection with contracts for the installation of high pressure and high temperature piping for central stations, industrial and process plants.

Prat-Daniel Corporation has appointed Frank Howell Company, Richmond, Va., as representative in that territory, handling Thermix dust collectors, fan stacks and air heaters.

Charles M. Chapman is now representing Cochrane Corporation on flow meters and The Hays Corporation on combustion instruments and automatic combustion control in the Cincinnati territory covering parts of Ohio, Indiana and Kentucky.

The Johnston & Jennings Co., Cleveland, Ohio, has announced the appointment of Robert H. Emerick as Assistant Sales Manager of their Stowe Stoker Division.

Electric Construction for 1940 Over $593,000,000

The Federal Power Commission reported on March 25 that the proposed construction expenditures by electric utilities during 1940 total $593,879,533. Of this amount $227,060,565 will be spent for generating plants and the remainder of $366,818,968 will be spent for other electric plant. Of the total, $490,434,261 are to be spent by privately owned utilities and the remainder, $103,445,272, will be spent by publicly owned utilities.

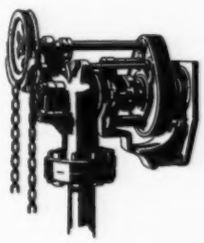
These expenditures will provide an additional installed capacity amounting to 1,877,844 kw during 1940, of which 1,332,635 kw are to be added by privately owned utilities and the remainder of 545,209 kw are to be installed by publicly owned utilities. The scheduled additions of hydroelectric installations by privately owned utilities total 57,000 kw and by publicly owned utilities 234,175 kw. Scheduled additions of 1,264,085 kw are to be added in privately owned steam plants and 270,372 kw in publicly owned steam plants; 11,550 kw of internal-combustion generating equipment will be added by privately owned utilities and 40,662 kw by publicly owned utilities.

On January 1, 1940, the total installed capacity of all generating equipment was 40,317,924 kw. The additions now under construction represent an increase of 4.7 per cent of the present installed capacity. The installed capacity of hydroelectric plants on January 1, 1940, was 11,415,165 kw, that of steam plants 28,046,948 kw and that of internal-combustion engine plants 355,811 kw. The proposed additions for 1940 represent increases of 2.6 per cent, 5.5 per cent and 6.1 per cent, respectively, of the hydro, steam and internal-combustion capacity.

Budgeted expenditures during the current year for hydroelectric plants total $3,337,110 by privately owned utilities and $46,731,000 by publicly owned utilities. Those for steam plants total $150,781,568 by privately owned utilities and $21,273,242 by publicly owned utilities. Similarly, budgeted expenditures for internal-combustion generating equipment total $1,098,967 by privately owned utilities and $3,838,678 by publicly owned utilities.

These figures cannot properly be used to determine the cost per kilowatt of the additional installed capacity, since a large portion of the 1,877,844 kw, scheduled for completion during 1940, was under construction prior to January 1 of this year and a portion of the construction expense has been covered by expenditures during previous years. Also some portion of the total expenditures listed will be spent for generating capacity which will not be placed in service until sometime subsequent to December 31, 1940.

In addition to the capacity scheduled for completion during 1940, privately owned electric utilities have 1,653,375 kw of additional capacity scheduled for completion subsequent to December 31, 1940, and publicly owned utilities have 758,100 kw scheduled for completion after that date.



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There are no patches of scale left at the bends after a Dean goes through. Vibratory action removes all deposit—outside slag as well as inside scale—and leaves the entire tube really clean.

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Write for Bulletin No. 133

The WM. B. PIERCE CO., 51 N. Division St., Buffalo, N. Y.

EQUIPMENT SALES
Boiler, Stoker, Pulverized Fuel

as reported by equipment manufacturers to the Department of Commerce, Bureau of the Census

Boiler Sales

	1940 Water Tube		1939 Water Tube		1940 Fire Tube		1939 Fire Tube	
	No.	Sq. Ft.	No.	Sq. Ft.	No.	Sq. Ft.	No.	Sq. Ft.
Jan.....	62	257,704	75	380,903	51	68,639	50	64,511
Feb.....	54	350,471	70	309,235	46	51,237	45	58,028
Jan.-Feb., Inclusive	116	608,175	145	690,138	97	119,876	95	122,539

Mechanical Stoker* Sales

	1940 Water Tube		1939 Water Tube		1940 Fire Tube		1939 Fire Tube	
	No.	Hp	No.	Hp	No.	Hp	No.	Hp
Jan.....	24	10,770	44	17,067	104	14,745	145	17,842
Feb.....	31	10,729	46	20,715	118	17,862	140	18,217
Jan.-Feb. Inclusive	55	21,499	90	37,782	222	32,607	285	36,059

* Capacity over 300 lb of coal per hour.

Pulverizer Sales

	1940 Water Tube			1939 Water Tube			1940 Fire Tube			1939 Fire Tube		
	No.		Lb Coal/- Hr	No.		Lb Coal/- Hr	No.		Lb Coal/- Hr	No.		Lb Coal/- Hr
	N.†	E.†		N.†	E.†		N.†	E.†		N.†	E.†	
Jan....	10	—	214,250	10	—	79,000	1	—	600	—	—	—
Feb.....	15	1	186,935	7	—	89,600	1	2	2,800	—	—	—
Jan.-Feb., Incl.	25	1	401,185	17	—	168,600	2	2	3,400	—	—	—

† N—New boilers; E—Existing boilers.

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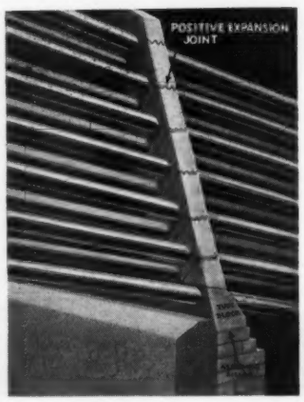

See these expansion joints

BECO-TURNER Baffles are distinguished by the expansion joints shown in the picture opposite. These joints afford relief from temperature changes, prevent cracking, and keep the baffles absolutely gas tight throughout their entire life. They also minimize damage to the baffle when removing warped tubes.

Beco-Turner Baffles are constructed of self-hardening refractory material which sets to concrete-like hardness. They are readily installed at any angle or in any shape necessary to bring about the most efficient heat transfer.

Ask for our big catalog covering Beco-Turner Baffles. Send blueprints showing your existing baffles for our recommendations and proposal.

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BECO-TURNER BAFFLES